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U. S. ARMY
TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

TCREC TECHNICAL REPORT 62-103

HOT CYCLE ROTOR DUCT CLOSURE VALVE SYSTEM

Task 1D121401D14403
(Formerly Task 9X38-01-020-03)

Contract DA 44-177-TC-832

March 1963

prepared by:

HUGHES TOOL COMPANY
Aircraft Division
Culver City, California

HEADQUARTERS
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
Fort Eustis, Virginia

Under the terms of Contract DA 44-177-TC-832, Hughes Tool Company, Aircraft Division, has conducted the detailed analysis, design and operation of a hot cycle rotor duct closure valve system for single-engine operation of the rotor system fabricated under Air Force Contract AF 33(600)-30271.

The conclusions presented in this report are concurred in by the U. S. Army Transportation Research Command, Fort Eustis, Virginia, the cognizant agency for the contract.

FOR THE COMMANDER:

KENNETH B. ABEL
Captain USA
Adjutant

APPROVED BY:

H. S. JOHNSON
USATRECOM Project Engineer

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TCREC Technical Report 62-103

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HOT CYCLE ROTOR DUCT CLOSURE VALVE SYSTEM

Report No. 62-32

Prepared by

Hughes Tool Company, Aircraft Division
Culver City, California

for

U. S. ARMY TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

Prepared by:

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FOREWORD

This report has been prepared by Hughes Tool Company - Aircraft Division under Army Contract DA 44-177-TC-832, Clause 2 Paragraph c, which requires a report containing the detailed analysis, design and operation of a Hot Cycle Rotor Duct Closure Valve System for single engine operation.

Design work presented in this report is a continuation of a previous preliminary study covering the need, possible designs, and location of duct closure valves. That study was made and reported in the "Hot Cycle Rotor System Engine-Rotor Control Study", Reference No. 1, prepared under Contract AF 33(600)-30271.

This work was performed at the Hughes Tool Company - Aircraft Division, Culver City, California, under the direction of Mr. H. O. Nay, Manager, Transport Helicopter Department, and under the direct supervision of Mr. J. L. Velazquez, Senior Project Engineer, Hot Cycle Program.

The Contract was effective on December 29, 1961. The work was completed on June 22, 1962.

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1. SUMMARY

A detailed study of various methods for changing the effective nozzle area of the hot cycle rotor system to provide acceptable single-engine operation has been made using preliminary concepts developed in Reference 1. Thermodynamic, aerodynamic, structural and economic considerations led to the selection of outboard duct closure valves mounted in both ducts adjacent to the blade tip cascade. A description of the mechanical design and operation is given, and the structural integrity is substantiated by a detailed stress analysis.

2. INTRODUCTION

The need of a blade duct closure valve for the hot cycle research aircraft was established in the "Hot Cycle Rotor System Engine-Rotor Control Study", Reference No. 1. The research aircraft, the preliminary design of which has been completed under the present Army Contract DA 44-177-TC-832, uses two gas generator versions of the General Electric T64 engine. In order to sustain flight with one engine inoperative, flow must be restricted to approximately 50 percent of the rotor tip nozzle area in order to achieve maximum power from the remaining operating engine. After studying several valve configurations, a design was selected that closes off a portion of each rotor blade nozzle by means of a pivoted vane at the outboard end of each duct. For single-engine operation, these vanes rotate and close off 50 percent of the total tip nozzle exhaust area. The following sections of this report contain the configuration selection, detailed design, operation, and stress analysis of these valves.

3. DUCT VALVE CONFIGURATION SELECTION

Section 4 of Reference 1 discusses the requirement and function of blade duct valves. It also discusses two general types of possible valve configurations; namely, (1) root valve, and (2) blade tip cascade closure valve, along with some of their advantages and disadvantages. It further states that a final choice of blade duct valve location must be based on further study.

Since the writing of Reference 1, a study has been conducted of both configurations with respect to their detail design, function, effect on propulsive efficiency, and effect on blade structural integrity. This study has resulted in the selection and final detail design of the blade tip cascade valve.

Initial thinking pointed toward the design of a valve to be installed in the blade root where the lower centrifugal g field would simplify the mechanical design. However, investigation showed that this type of installation would give rise to structural and performance problems in the blade when the inboard valve was actuated to close off one duct. The structural problems arose from stresses produced by differential pressure and temperature considerations. Static pressure in the active duct of approximately 24 psig combines with approximately -6 psig in the closed off duct to produce in the order of 30 psig pressure difference between the two ducts. The negative pressure in the closed off duct results from centrifugal pumping and tip vortex negative pressure field. This 30-psig pressure difference would overstress the structure separating the ducts. An additional overstress results from thermal stresses produced by the temperature difference between the ducts of approximately 1000 degrees Fahrenheit. In order to solve these structural problems, substantial redesign and modification work would be required on the present rotor at a cost that would be unacceptable within the funding limitations of this project.

From the thermodynamic efficiency point of view, it has been determined that allowing both ducts to function during single-engine operation yields 35 percent more available power than that available by restricting the flow to one duct. This improvement results from the elimination of leakage from one duct to the other and the lower friction losses associated with using both ducts for the gas flow from one engine.

Summarizing, the selected design of the blade tip cascade valves, by closing off one-half of each cascade exit and allowing equal temperatures, pressures, and mass flow to exist in each duct, eliminates unacceptable structural and thermodynamic features of the root installation. Effects of the higher centrifugal forces at the tip on design complexity are offset by improved accessibility and simplified work required for modification to the existing blades.

4. MECHANICAL DESIGN

The blade tip cascade valves will be either fully closed or open, depending upon whether one engine or two engines are operating. Their functioning will be automatic with manual override. A pressure sensor will compare total engine discharge pressures. Following a pressure drop due to engine failure, an electrical actuator will be energized which will operate the linkage to close the valves. The pilot can, by manual switch selection, return the valves to the open position when dual-engine operation has been restored. The pilot will also have the capability of closing the valves in the event of the malfunction of the automatic system.

Mechanical design of the rotor blade duct closure valves, as located close to the tip cascade, is influenced by: (a) a high g field (857 g max.), (b) a high operating temperature (1200 degrees Fahrenheit), and (c) the necessity for a seal to operate at approximately 30 psi. The following drawings show the layout and details of the valves.

Figure	Drawing
1	Layout-Cascade Valve
2	Tip Assembly, Sheets 1 and 2
3	Forward Duct Valve
4	Aft Duct Valve
5	Forward Duct Sector
6	Aft Duct Lever

With an operating temperature of 1200 degrees Fahrenheit, a nickel base high-temperature alloy, René 41, was selected for use for almost all components. This metal has performed quite well in the hot cycle rotor ducting system. (See Reference 2). René 41 corrosion resistant steel in the aged condition has an ultimate strength of 194,000 psi, and a yield strength of 145,000 psi at 1200 degrees Fahrenheit. In addition to high strength, using the same metal throughout minimizes the possibility of interference of moving parts due to thermal expansion. From recent experience with the hot cycle rotor sealing problems as reported in Reference 2, it has been found that heat cycling tends to cause warpage of most metals. This action could also cause interference or seizure of moving parts. To minimize this problem, the room temperature clearances of all moving parts is made as large as possible.

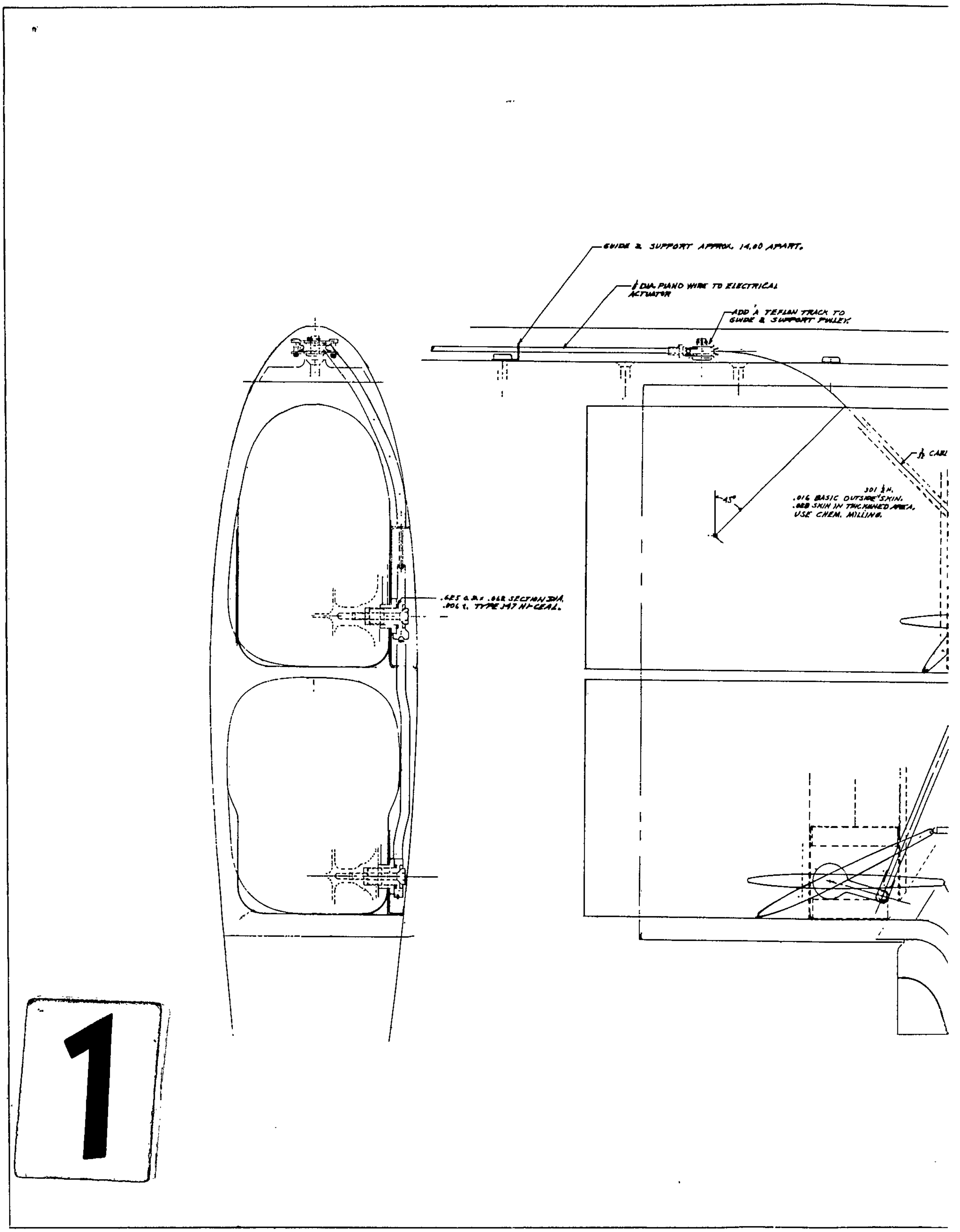
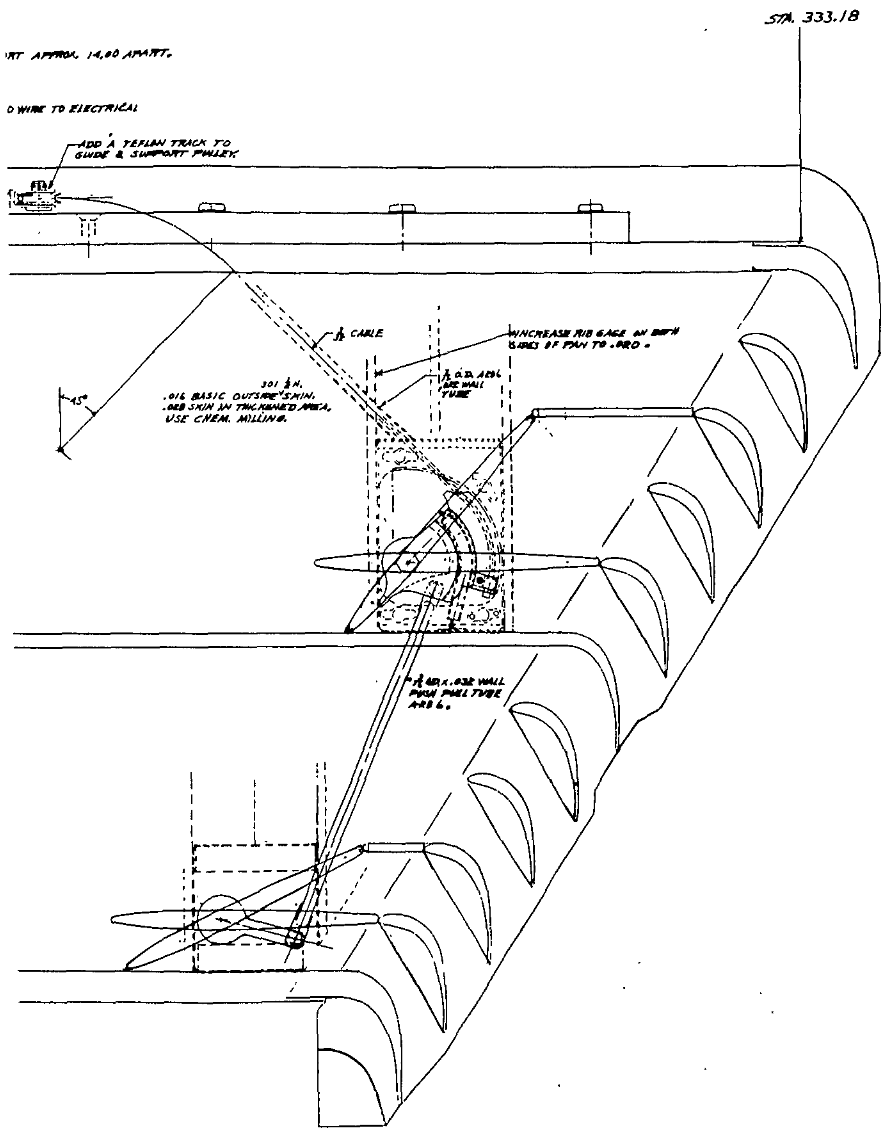


Figure 1. Layout-Cascade Valve.



2

REQ'D	PART NO.	REQ'D	PART NO.	NAME	SIZE	DESCRIPTION	SPECIFICATION
ASSEMBLY OPP.		ASSEMBLY SHOWN		LIST OF MATERIAL			

NEXT ASSY | USED ON | QTY REQD | APPLICATION

DRWN Sellows | CHK'D | APP'D | APP'D | APP'D | APP'D | APP'D | APP'D

LAYOUT-
CASCADE VALVE

SCALE FULL

HUGHES TOOL CO.
AIRCRAFT DIVISION
CULVER CITY, CALIF.

385-1102

CODE 02731

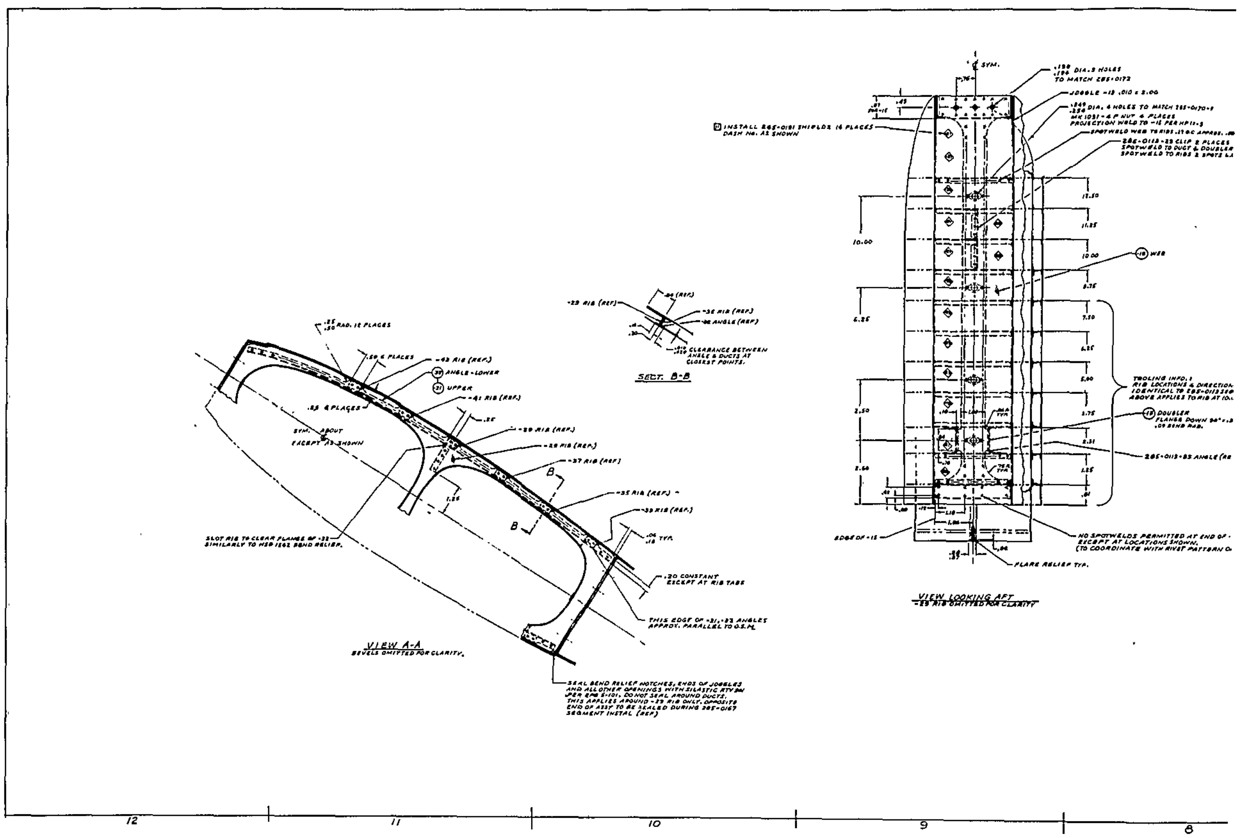
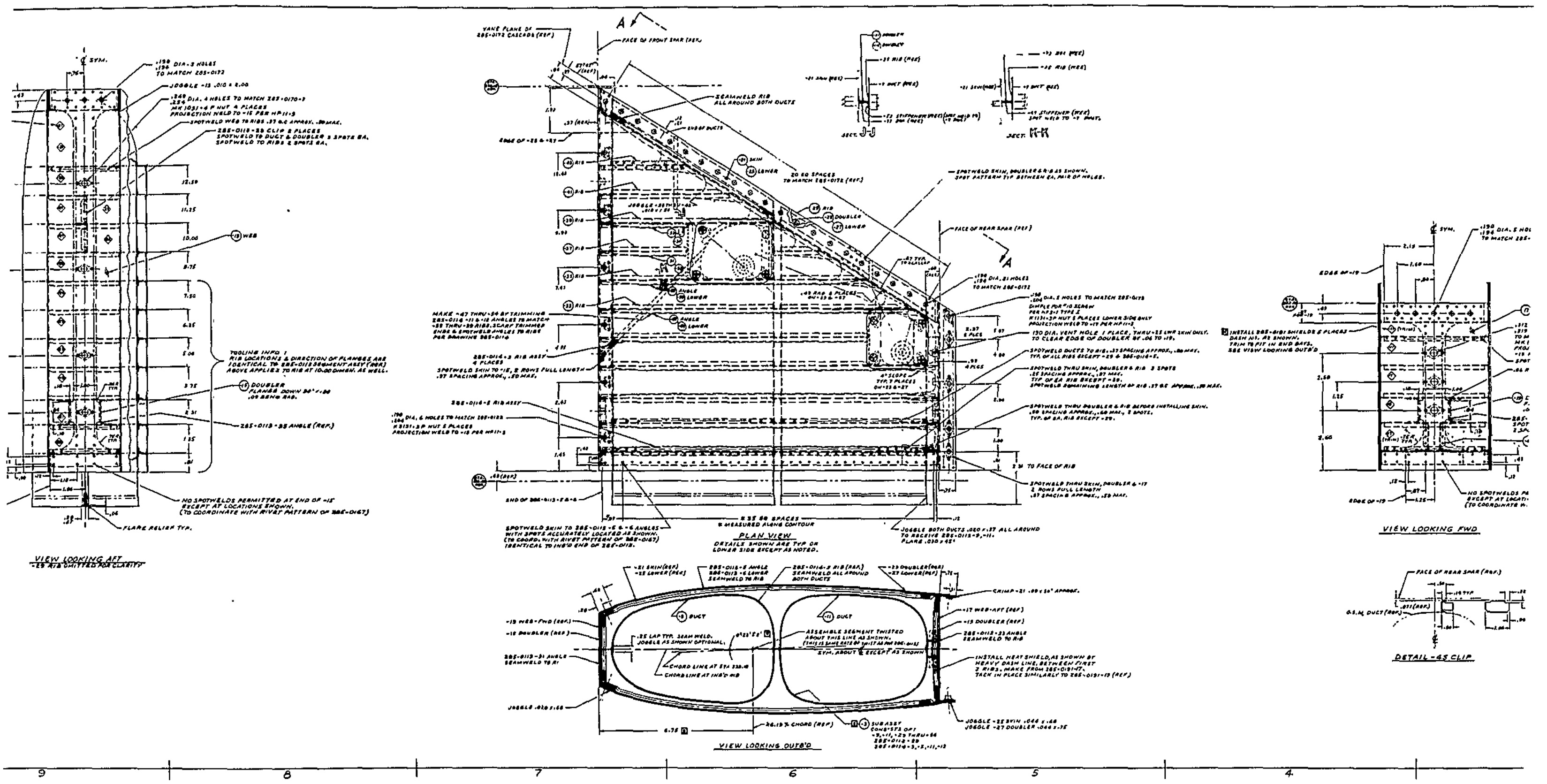
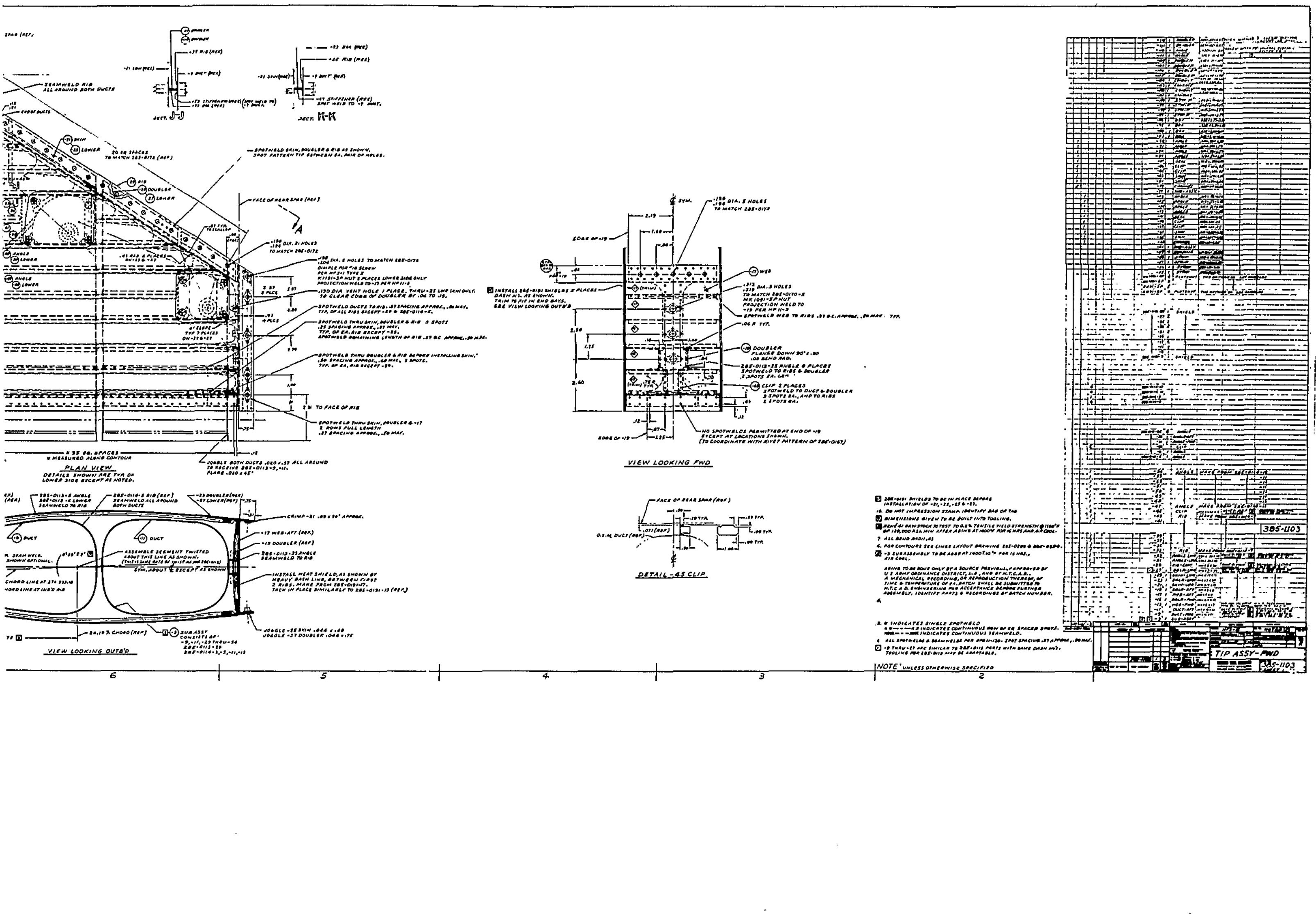


Figure 2a. Tip Assembly, Sheet 1.

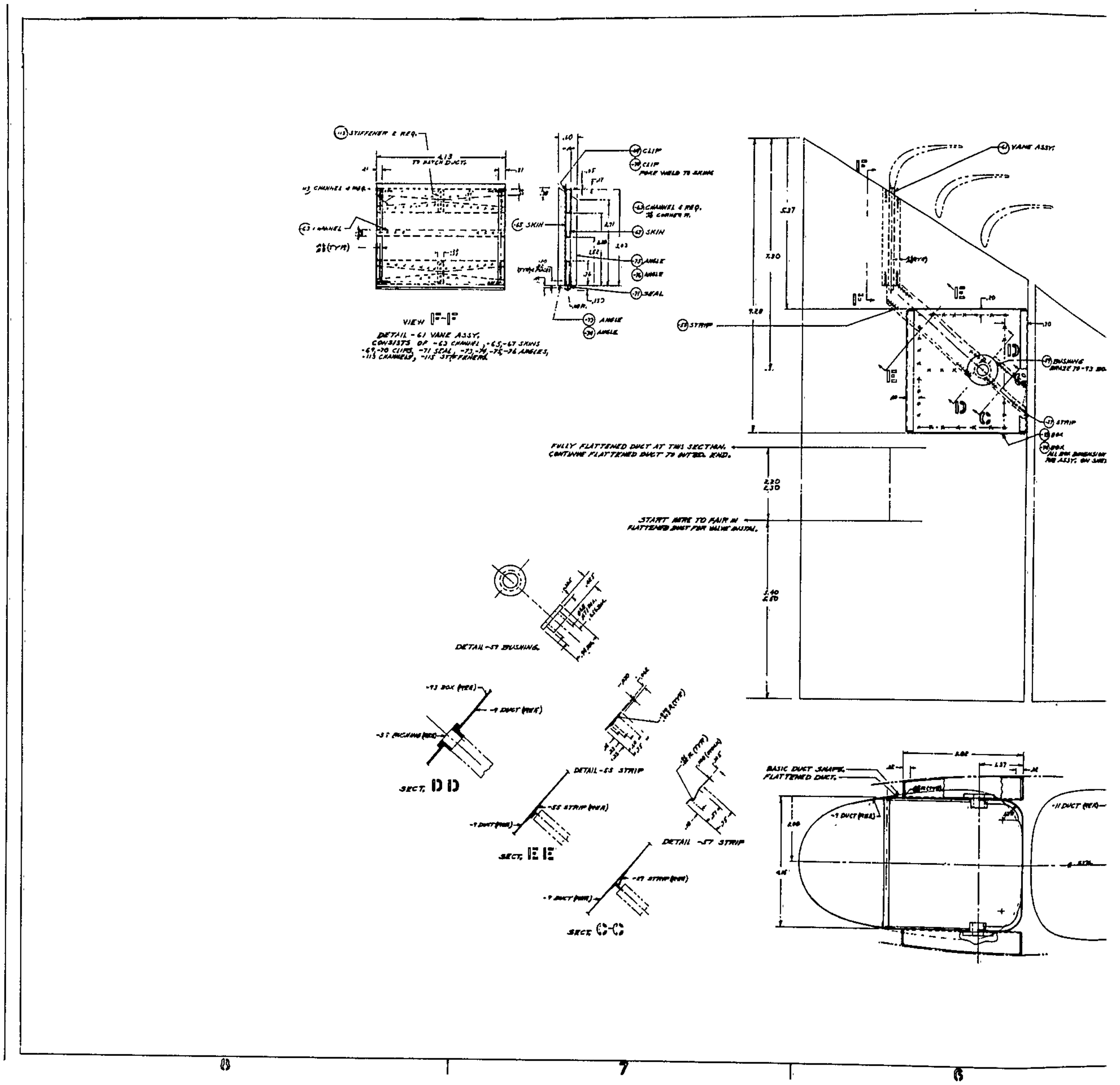

VIEW LOOKING AFT
PLAN VIEW
VIEW LOOKING OUTB'D
VIEW LOOKING FWD
DETAIL -45 CLIP
SECT. J-J
SECT. K-K
9
8
7
6
5
4

2



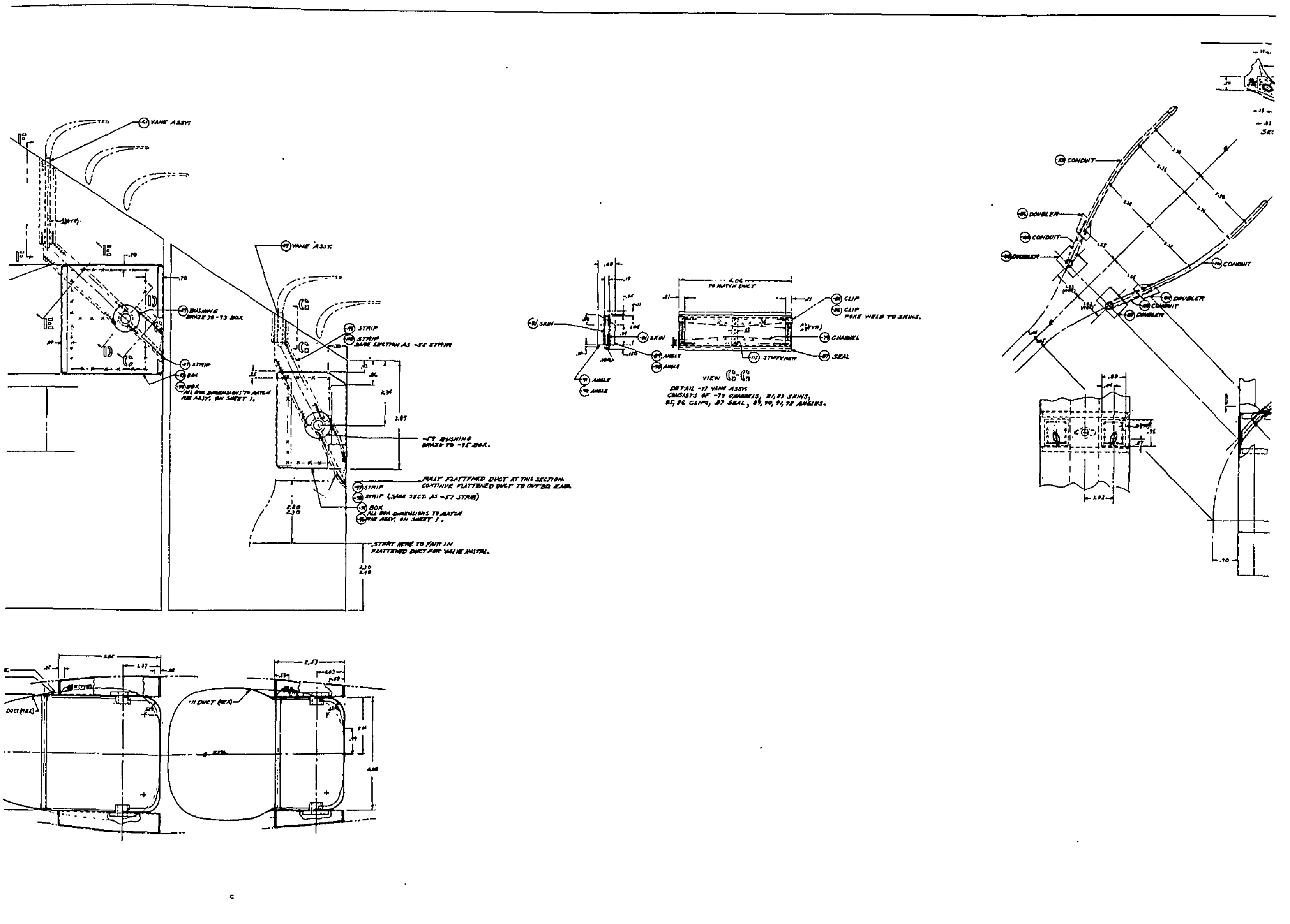
SECT. J-J
SECT. K-K
PLAN VIEW
DETAILS SHOWN ARE TYP. OF LOWER SIDE EXCEPT AS NOTED.
VIEW LOOKING OUTB'D
VIEW LOOKING FWD
DETAIL -45 CLIP
ALL BEND RADII .03
FOR CONTOURS SEE LINES LAYOUT DRAWING
NOTE: UNLESS OTHERWISE SPECIFIED
385-1103
TIP ASSY-FWD
385-1103

3



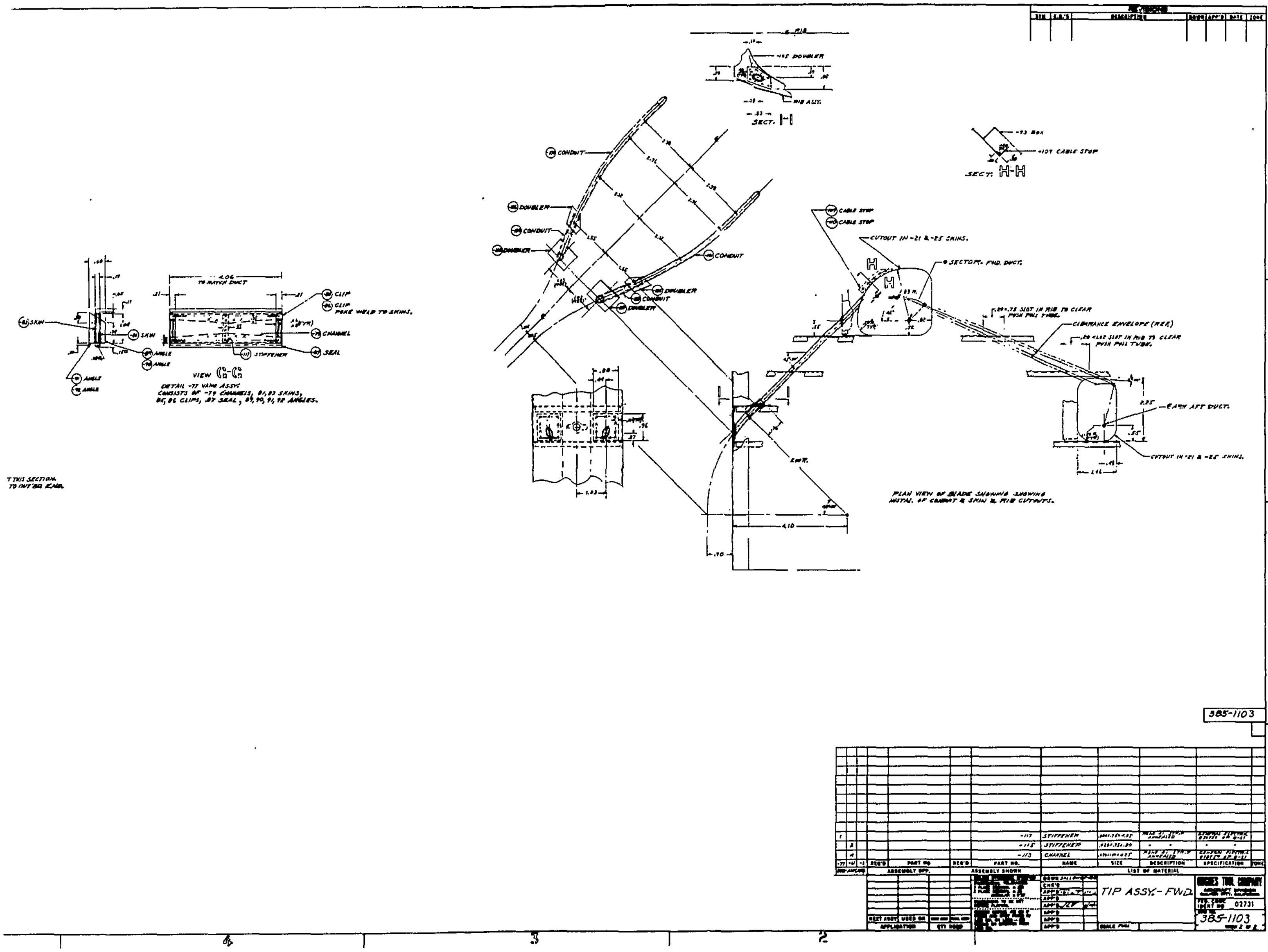
1

Figure 2b. Tip Assembly, Sheet 2.


VANE ASSY.
BUSHING
BRAZE TO -73 BOX
STRIP
BOX
ALL BOX DIMENSIONS TO MATCH RIB ASSY. ON SHEET 1.
VANE ASSY.
STRIP
STRIP
SAME SECTION AS -55 STRIP
-59 BUSHING
BRAZE TO -75 BOX.
FULLY FLATTENED DUCT AT THIS SECTION.
CONTINUE FLATTENED DUCT TO OUT'BD END.
STRIP
STRIP (SAME SECT. AS -57 STRIP)
BOX
ALL BOX DIMENSIONS TO MATCH RIB ASSY. ON SHEET 1.
START HERE TO FAIR IN FLATTENED DUCT FOR VALVE INSTAL.
SKIN
SKIN
ANGLE
ANGLE
ANGLE
ANGLE
4.06
TO MATCH DUCT
CLIP
CLIP
POKE WELD TO SKINS.
CHANNEL
STIFFENER
SEAL
VIEW
DETAIL -77 VANE ASSY.
CONSISTS OF -79 CHANNELS, 81, 83 SKINS,
85, 86 CLIPS, 87 SEAL, 89, 90, 91, 92 ANGLES.
CONDUIT
DOUBLER
CONDUIT
DOUBLER
CONDUIT
DOUBLER
CONDUIT
DOUBLER
-11 DUCT (REF)
DUCT (REF)

6 | 5 | 4 | 3

2


SECT. I-I
SECT. H-H
VIEW G-G
DETAIL -77 VANE ASSY. CONSISTS OF -79 CHANNELS, 81, 83 SKINS, 85, 86 CLIPS, 87 SEAL, 89, 90, 91, 92 ANGLES.
PLAN VIEW OF BLADE SHOWING SHOWING INSTAL. OF CONDUIT & SKIN & RIB CUTOUTS.
CUTOUT IN -21 & -25 SKINS.
CLEARANCE ENVELOPE (REF.)
CONDUIT
DOUBLER
CABLE STOP
CLIP
CHANNEL
SEAL
STIFFENER
ANGLE
SKIN
385-1103
-117 STIFFENER
-115 STIFFENER
-113 CHANNEL
PART NO.
NAME
SIZE
DESCRIPTION
SPECIFICATION
ASSEMBLY OPP.
ASSEMBLY SHOWN
LIST OF MATERIAL
NEXT ASSY
USED ON
APPLICATION
QTY REQD
DRWN
CHK'D
APP'D
TIP ASSY.- FWD.
HUGHES TOOL COMPANY
02731
385-1103
SCALE FULL

3

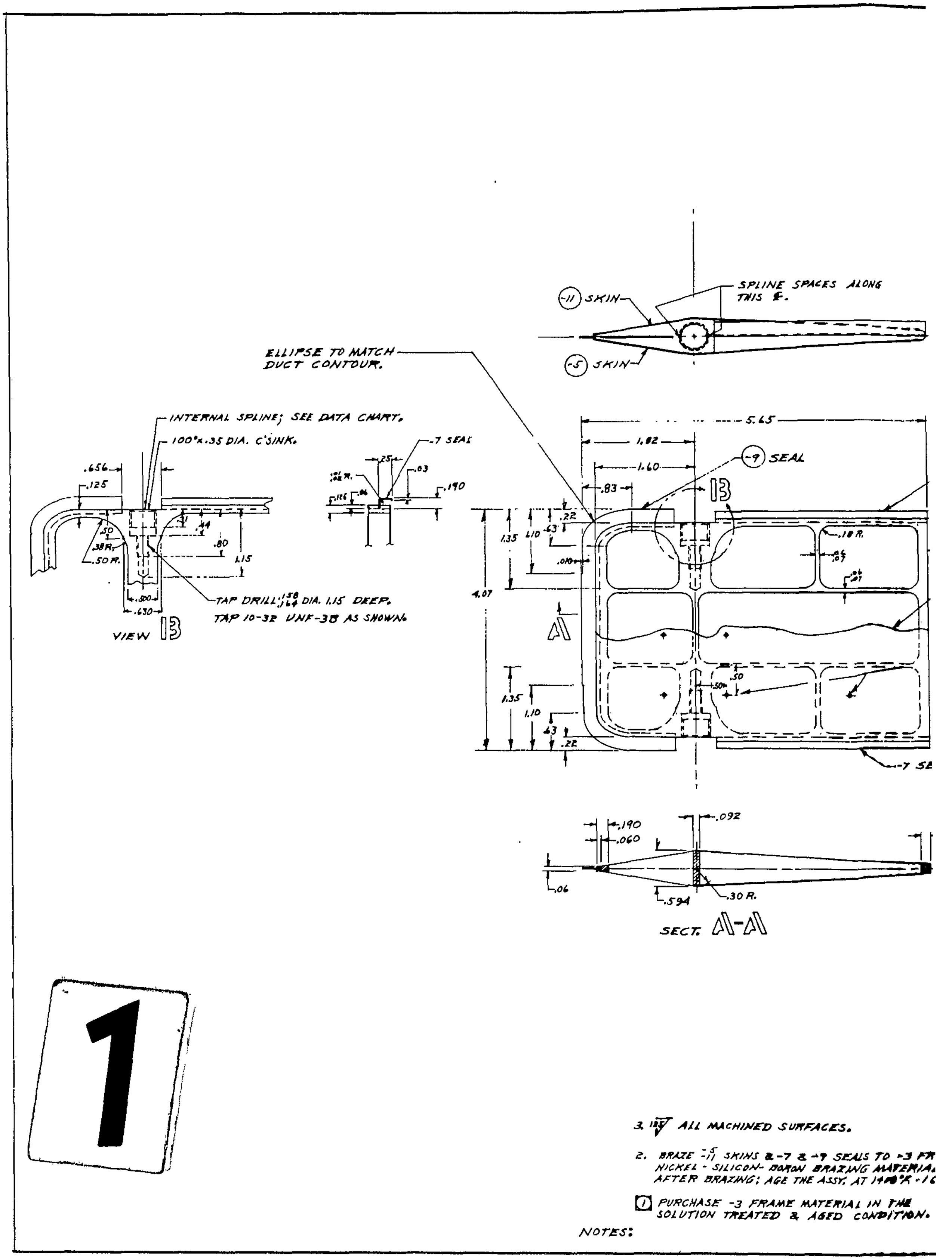
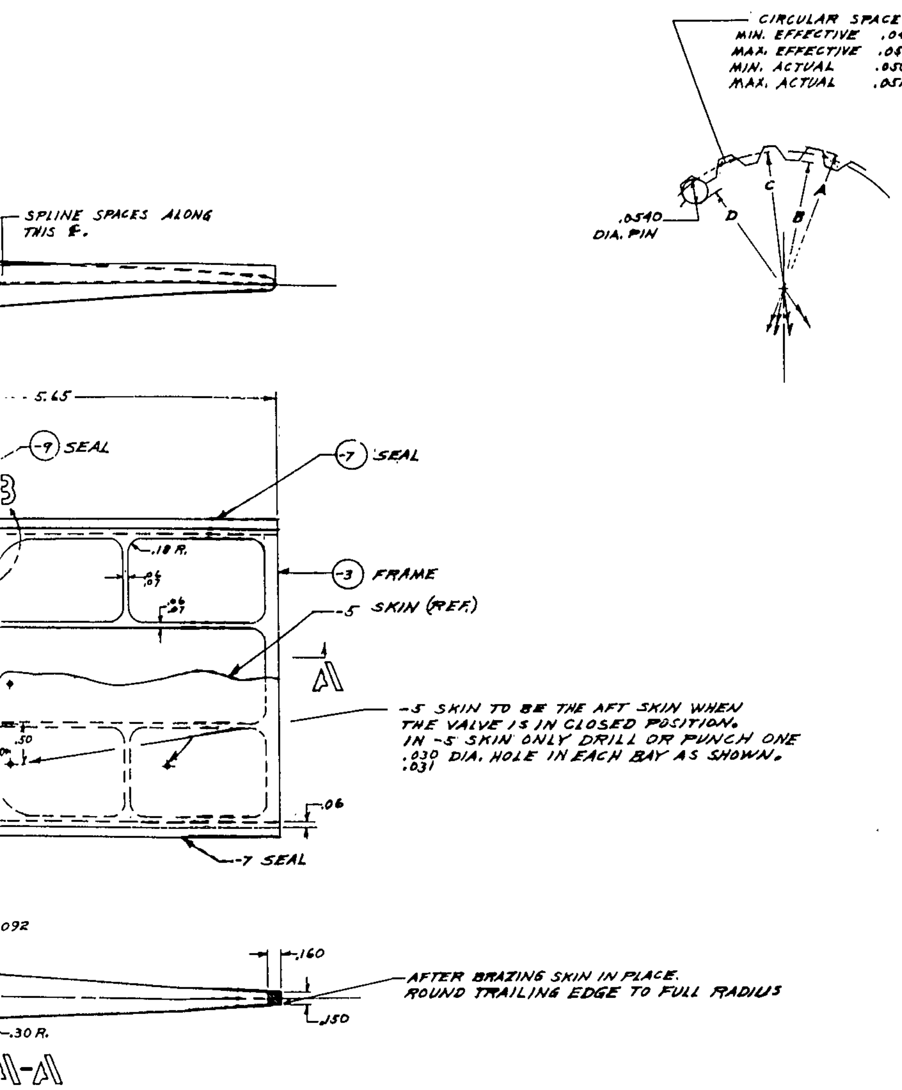


Figure 3. Forward Duct Valve.

SYM	E.O.'S	DESCRIPTION	DRWN	APP'D	DATE

(REVISIONS)



INTERNAL SPLINE DATA

FLAT ROOT SIDE FIT	
NO. OF TEETH	14
PITCH	32/64
PRESSURE ANGLE	30°
A MAJOR DIA.	.4688-.4718
B MINOR DIA.	.4081-.4111
C PITCH DIA.	.4375
D MEASUREMENT BETWEEN PINS MAX.	.3634 (REF.)
125 ALL SPLINE SURFACES	

385-1104

...ACHINED SURFACES.

...SKINS & -7 & -9 SEALS TO -3 FRAME WITH ...ILICON- BORON BRAZING MATERIAL. ...AZING: AGE THE ASSY. AT 1400°F - 16 HRS, AIR COOL.

... -3 FRAME MATERIAL IN THE ... TREATED & AGED CONDITION.

REQD	PART NO.	NAME	SIZE	DESCRIPTION	SPECIFICATION
1	-11	SKIN	.010×4.50×6.00	RENE 41 STRIP ANNEALED	GENERAL ELECTRIC B50T59-S2
1	-9	SEAL	.006×3.10×5.00	RENE 41 STRIP ANNEALED	GENERAL ELECTRIC B50T59-S2
2	-7	SEAL	.006×.60×4.00	" "	" "
1	-5	SKIN	.010×4.50×6.00	RENE 41 STRIP ANNEALED	GENERAL ELECTRIC B50T59-S2
1	-3	FRAME	.500×4.50×6.00	RENE 41 PLATE	" " "

ASSEMBLY SHOWN — LIST OF MATERIAL

REQD	PART NO. (ASSEMBLY OPP.)
	①

NEXT ASSY	USED ON	QTY REQD NEXT ASSY	QTY REQD FINAL ASSY
385-1100	385-1000	1	3

APPLICATION

UNLESS OTHERWISE SPECIFIED
DIMENSIONAL TOLERANCES
3 PLACE DECIMAL ± .010
2 PLACE DECIMAL ± .03
ANGULAR ± 0°30'
DIMENSIONS TO BE MET BEFORE PLATING.
CORNER RADIUS .062 ON C' BORES AND SPOT FACES OF 1.250 DIA. OR LESS — .093 RADIUS ON GREATER THAN 1.250 DIA.

DRWN	SALLOWS	4-20-62
CHK'D		
APP'D		6-13-62
APP'D		
APP'D	JLV	6/14/62
APP'D		
APP'D		
APP'D		

CASCADE VALVE — FORWARD DUCT.

SCALE FULL

HUGHES TOOL COMPANY
AIRCRAFT DIVISION
CULVER CITY, CALIFORNIA

385-1104

CODE 02731 SHEET OF

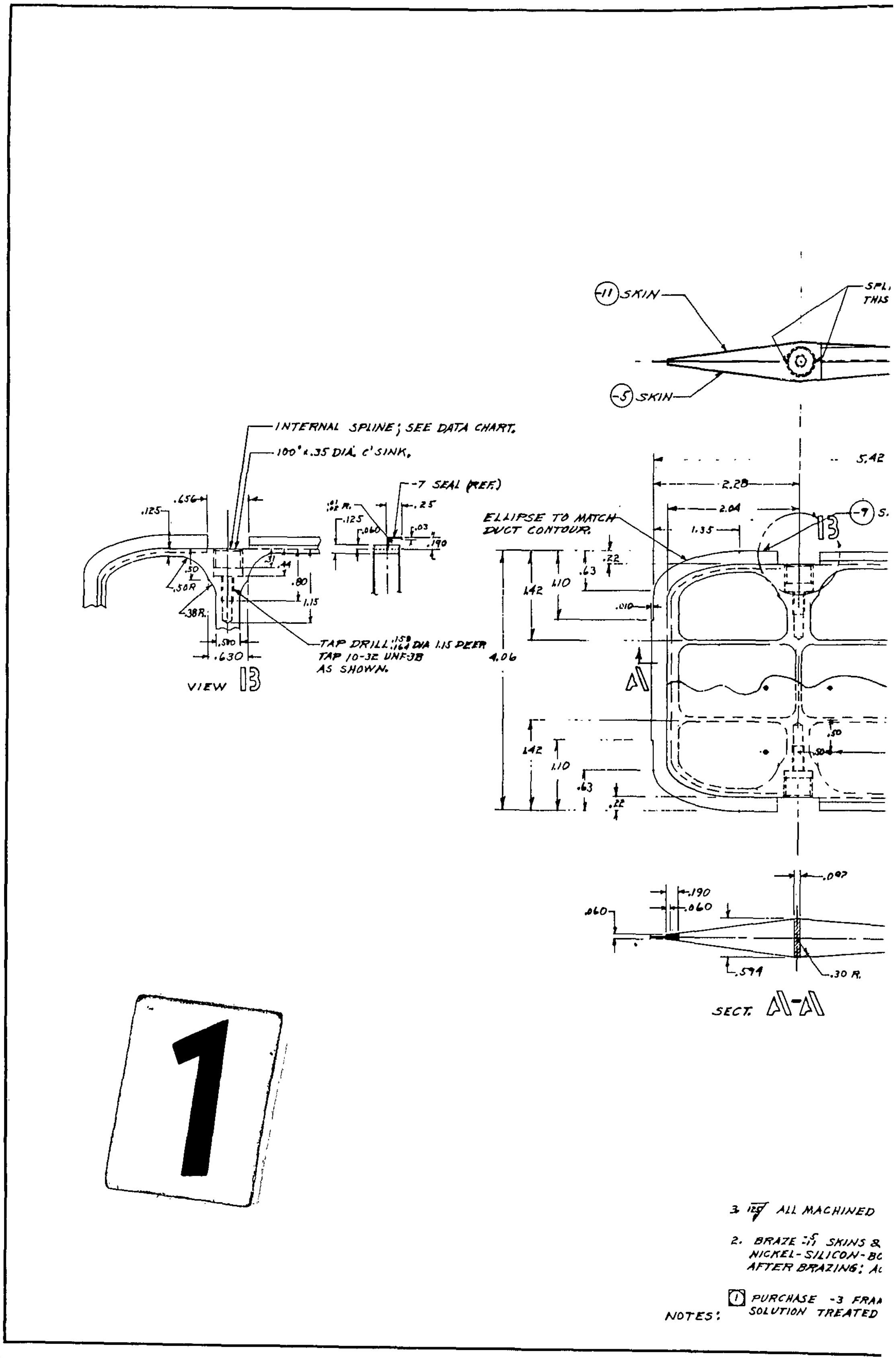
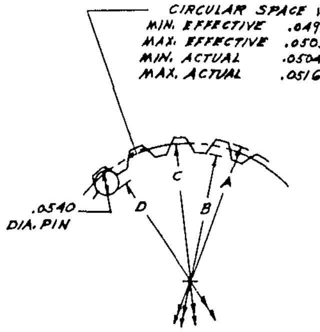
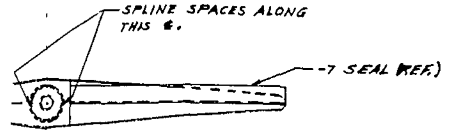


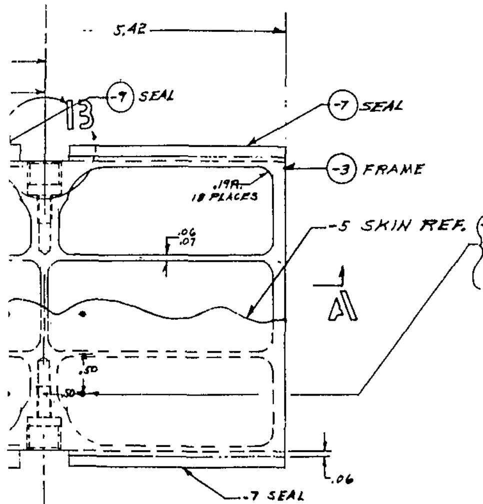
Figure 4. Aft Duct Valve.

REVISIONS					
SYM	E.O.'S	DESCRIPTION	DRWN	APP'D	DAT

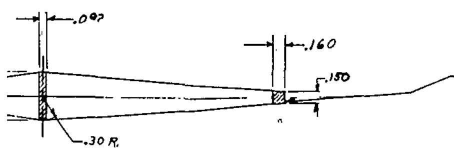


INTERNAL SPLINE DATA	
FLAT ROOT SIDE FIT	
NO. OF TEETH	14
PITCH	32/64
PRESSURE ANGLE	30°
A MAJOR DIA.	.4688-.4718
B MINOR DIA.	.4081-.4111
C PITCH DIA.	.4375
D MEASUREMENT BETWEEN PINS MAX.	.3634 (REF)
ALL SPLINE SURFACES.	





-5 SKIN TO BE THE AFT SKIN WHEN THE VALVE IS IN CLOSED POSITION.
IN -5 SKIN ONLY DRILL OR PUNCH ONE .030/.031 DIA. HOLE IN EACH BAY AS SHOWN.



AFTER BRAZING SKIN IN PLACE, ROUND TRAILING EDGE TO FULL RADIUS

A-A

2

385-1107

REQD	PART NO.	REQD	PART NO.	NAME	SIZE	DESCRIPTION	SPECIFICATION
		1	-11	SKIN	.010×4.50×6.00	RENE 41 STRIP ANNEALED	GENERAL ELECTRIC B50T59-S2
		1	-9	SEAL	.006×3.00×5.00	RENE 41 STRIP ANNEALED	GENERAL ELECTRIC B50T59-S2
		2	-7	SEAL	.006×.60×3.00	" "	" "
		1	-5	SKIN	.010×4.50×6.00	RENE 41 STRIP ANNEALED	GENERAL ELECTRIC B50T59-S2
	(1)	1	-3	FRAME	.500×4.50×6.00	RENE 41 PLATE	" " "
ASSEMBLY OPP.		ASSEMBLY SHOWN		LIST OF MATERIAL			

... ALL MACHINED SURFACES.

...RAZE -5/-11 SKINS & -7 & -9 SEALS TO -3 FRAME WITH ...CKEL-SILICON-BORON BRAZING MATERIAL.
...TER BRAZING: AGE THE ASSY. AT 1400°F-16 HRS., AIR COOL.

...URCHASE -3 FRAME MATERIAL IN THE ...LUTION TREATED & AGED CONDITION.

NEXT ASSY	USED ON	NEXT ASSY	FINAL ASSY
385-1100	385-1000	1	3
APPLICATION		QTY REQD	

UNLESS OTHERWISE SPECIFIED
DIMENSIONAL TOLERANCES
3 PLACE DECIMAL ± .010
2 PLACE DECIMAL ± .03
ANGULAR ± 0°30'
DIMENSIONS TO BE MET BEFORE PLATING.
CORNER RADIUS .062 ON C' BORES AND SPOT FACES OF 1.250 DIA. OR LESS — .093 RADIUS ON GREATER THAN 1.250 DIA.

DRWN	SALLOWS	5-1-62
CHK'D		
APP'D	STRESS [illegible]	6-11-62
APP'D		
APP'D	JZV	6/14/62
APP'D		
APP'D		
APP'D		

CASCADE VALVE - AFT DUCT

SCALE FULL

HUGHES TOOL COMPAN
AIRCRAFT DIVISION
CULVER CITY, CALIFORNIA

385-1107

CODE 02731 SHEET OF

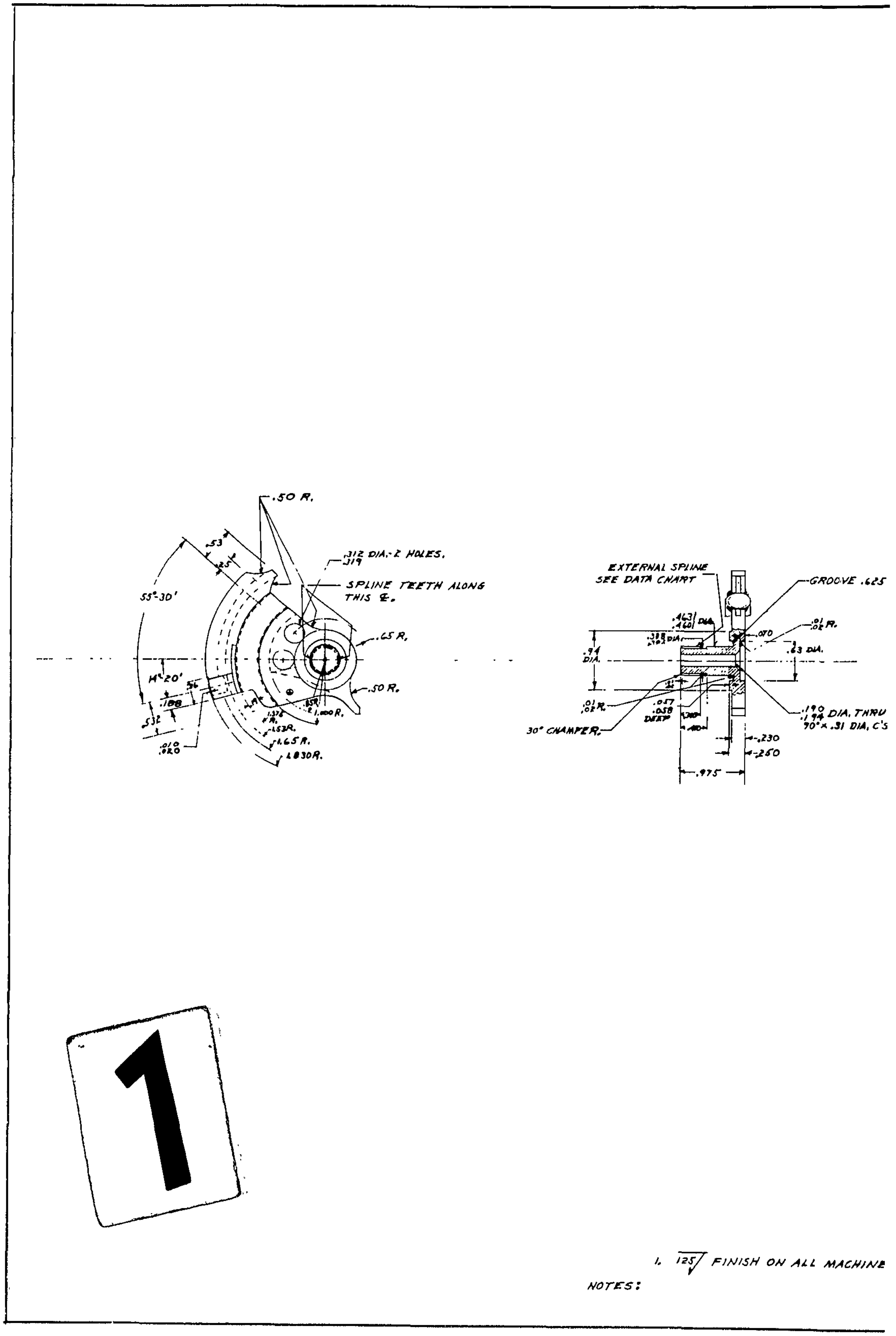
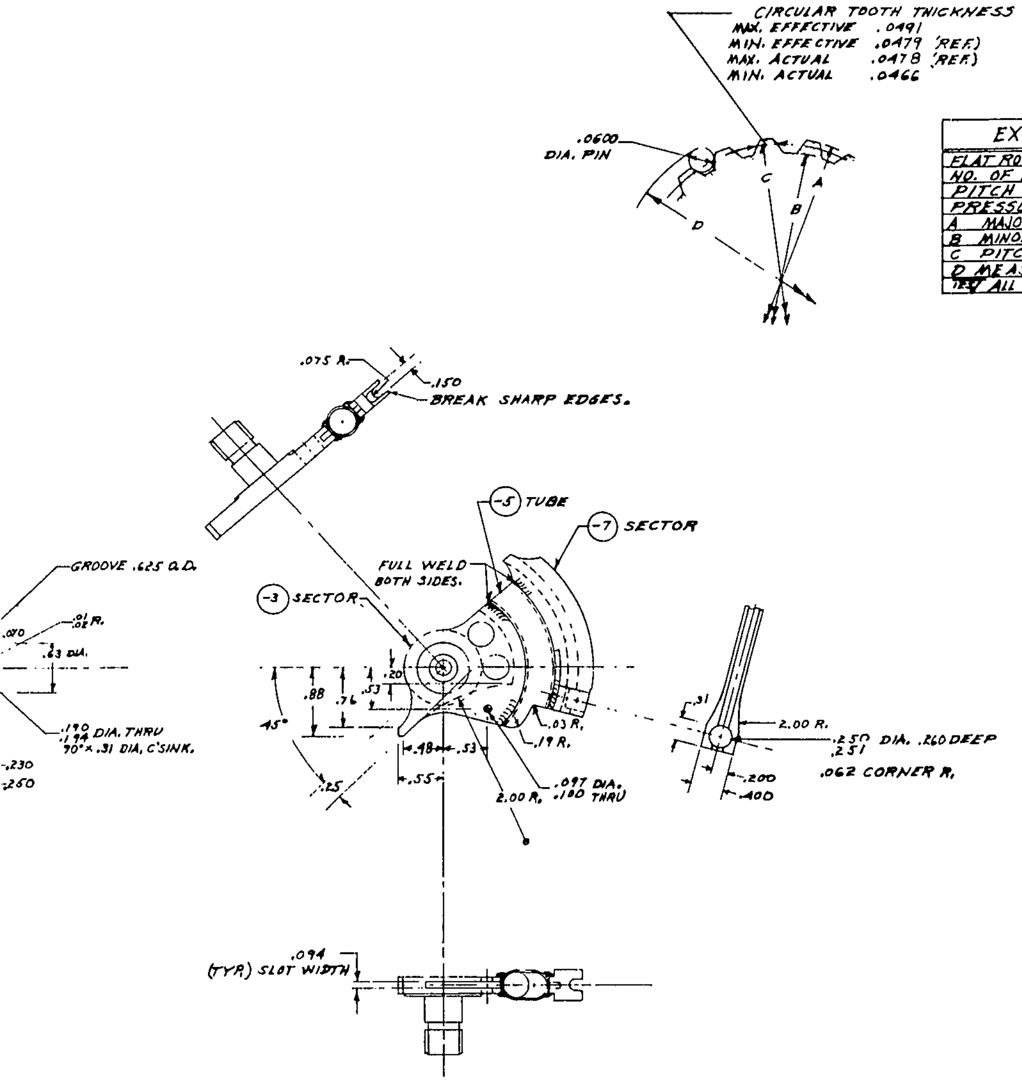


Figure 5. Forward Duct Sector.

REVISIONS					
SYM	E.O.'S	DESCRIPTION	DRWN	APP'D	DATE



EXTERNAL SPLINE DATA	
FLAT ROOT SIDE FIT	
NO. OF TEETH	14
PITCH	32/64
PRESSURE ANGLE	30°
A MAJOR DIA.	.463 -.461
B MINOR DIA.	.3960-.3880
C PITCH DIA.	.4375
D MEASUREMENT OVER PINS	MIN. .5379(REF)
TEST ALL SPLINE SURFACES	

2

385-1106

H ON ALL MACHINE SURFACES.

REQD	PART NO.	REQD	PART NO.	NAME	SIZE	DESCRIPTION	SPECIFICATION
1	-7	1	-7	SECTOR	1.00×1.50×3.00	RENE 41 BAR SOLUTION TREATED & AGED.	GENERAL ELECTRIC B 50T59-52
1	-5	1	-5	TUBE	.375 O.D. × .032 WALL × 2.50 LONG	RENE 41 TUBE ANNEALED	" "
1	-4	1	-3	SECTOR	1.00×2.00×2.00	RENE 41 BAR SOLUTION TREATED & AGED.	" "
-2 ASSEMBLY OPP.		-1 ASSEMBLY SHOWN		LIST OF MATERIAL			

NEXT ASSY	USED ON	NEXT ASSY	FINAL ASSY
385-1100	385-1000	1 OF -2 1 OF -1	3 OF -2 3 OF -1
APPLICATION		QTY REQD	

UNLESS OTHERWISE SPECIFIED
DIMENSIONAL TOLERANCES
3 PLACE DECIMAL ± .010
2 PLACE DECIMAL ± .03
ANGULAR ± 0°30'
DIMENSIONS TO BE MET BEFORE PLATING.
CORNER RADIUS .062 ON C' BORES AND SPOT FACES OF 1.250 DIA. OR LESS — .093 RADIUS ON GREATER THAN 1.250 DIA.

DRWN	SALLOWS	4-26-62
CHK'D		
APP'D	[illegible]	6-13-62
APP'D		
APP'D	JZV	6/14/62
APP'D		
APP'D		
APP'D		

SECTOR - FORWARD DUCT CASCADE VALVE.

SCALE FULL

HUGHES TOOL COMPANY
AIRCRAFT DIVISION
CULVER CITY, CALIFORNIA

385-1106

CODE 02731 SHEET OF

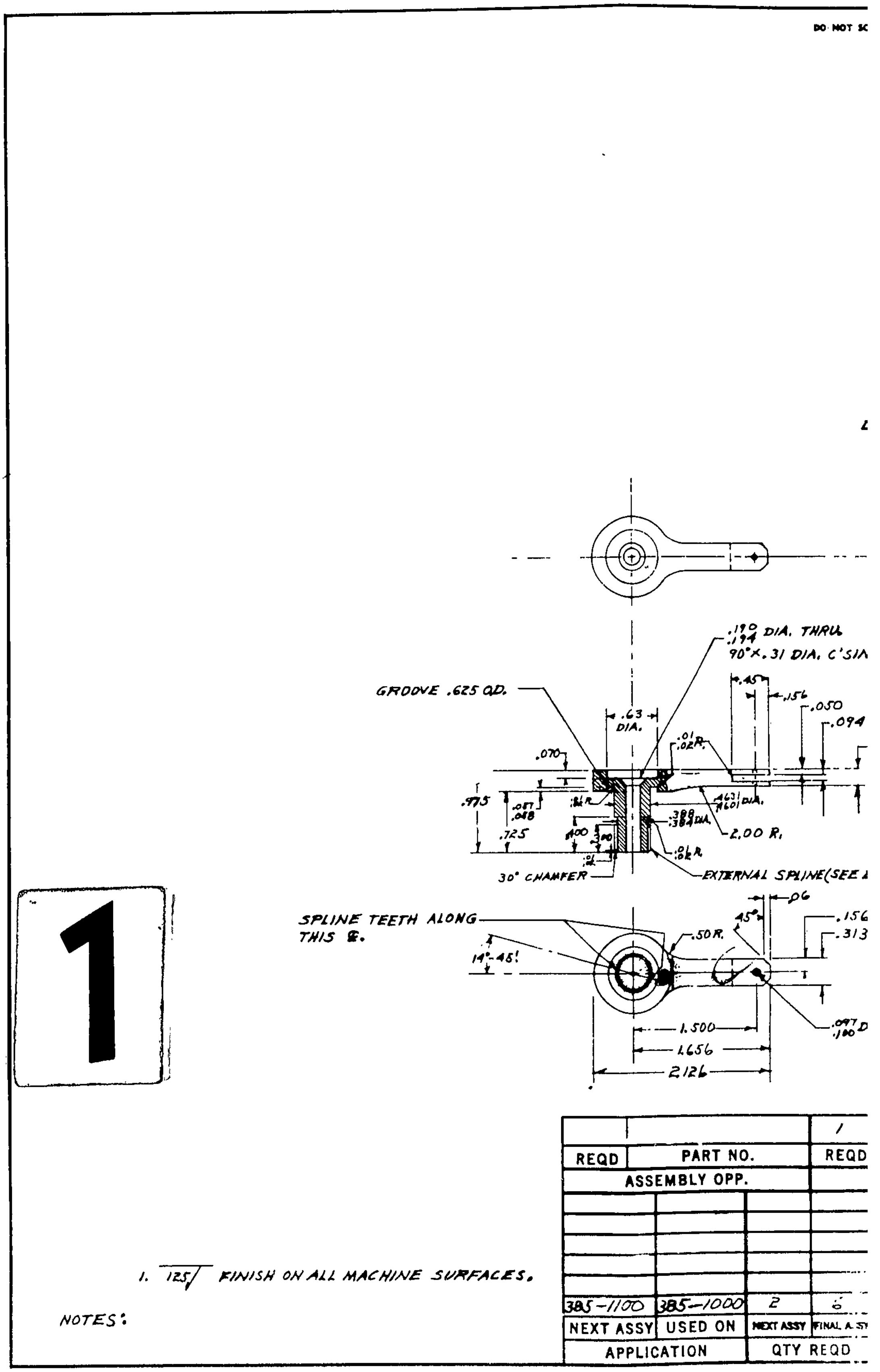
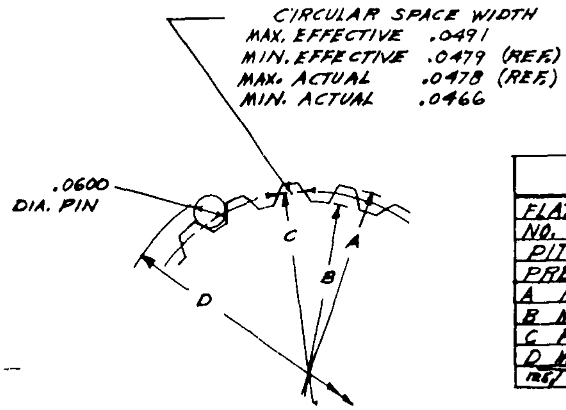


Figure 6. Aft Duct Lever.

DO NOT SCALE

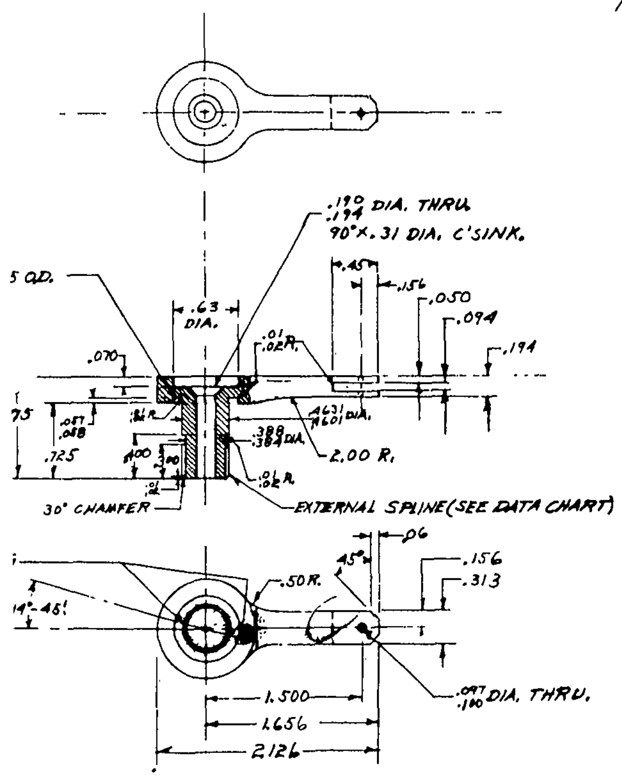
REVISIONS

SYM	E.O.'S	DESCRIPTION	DRWN	APP'D	DATE



EXTERNAL SPLINE DATA	
FLAT ROOT SIDE FIT	
NO. OF TEETH	14
PITCH	32/64
PRESSURE ANGLE	30°
A MAJOR DIA.	.463 -.461
B MINOR DIA.	.3960-.3880
C PITCH DIA.	.4375
D MEASUREMENT OVER PINS.	MIN. .5379 (REF.)
(ref) ALL SPLINE SURFACES	

385-1105



2

		1	-3	LEVER	1.00X1.50X3.00	RENE 41 BAR SOLUTION TREATED & AGED	GENERAL ELECTRIC B50T59-52
REQD	PART NO.	REQD	PART NO.	NAME	SIZE	DESCRIPTION	SPECIFICATION
ASSEMBLY OPP.		ASSEMBLY SHOWN		LIST OF MATERIAL			

NEXT ASSY	USED ON	NEXT ASSY	FINAL ASSY				
385-1100	385-1000	2	6				
APPLICATION		QTY REQD					

UNLESS OTHERWISE SPECIFIED
DIMENSIONAL TOLERANCES
3 PLACE DECIMAL ± .010
2 PLACE DECIMAL ± .03
ANGULAR ± 0°30'
DIMENSIONS TO BE MET BEFORE PLATING.
CORNER RADIUS .062 ON C' BORES AND SPOT FACES OF 1.250 DIA. OR LESS — .093 RADIUS ON GREATER THAN 1.250 DIA.

DRWN	SALLOWS	5-2-62
CHK'D		
APP'D STRESS	[illegible]	6-13-62
APP'D		
APP'D	[illegible]	[illegible]
APP'D		
APP'D		
APP'D		

LEVER - AFT DUCT CASCADE VALVE

HUGHES TOOL COMPANY
AIRCRAFT DIVISION
CULVER CITY, CALIFORNIA

385-1105

SCALE

CODE 02731 SHEET OF

...t Lever.

Before proceeding too far into the actual valve design, a study was made of the pressure loading involved. These loads are reported in the aerodynamic section of this report. As stated on page 31, the valve pivot point is located a short distance aft of the aerodynamic center and this location creates an unstable loading condition during the initial starting period. To correct this instability, springs are added to the forward duct sector (Figure 3) to hold both forward and aft valves in a steady state open position.

The valves, Figures 3 and 4, are fabricated from a machined framework covered with an 0.10-inch skin which is brazed to the framework. (Along the upper and lower edge and leading edge, a 0.006-inch thick lip-type seal is brazed in place. All seals are designed so that gas pressure aids in the sealing action.) A nickel-silicon-boron brazing material has been selected which becomes a liquid at approximately 1900 degrees Fahrenheit. After brazing, the assembly is aged at 1400 degrees Fahrenheit for 16 hours and air cooled to give maximum strength.

As mentioned previously, the valves are actuated by electrical actuators located at the inboard end of the blade. Present plans, formulated for expediency and lower cost advantages, call for the off-the-shelf purchase of electrical actuators to operate from a 28-volt DC supply source, have a travel of ±3, and exert a force of 680 pounds $^{-0}_{+25}$ percent. These actuators will not be irreversible and will be self-locking, thus preventing the valves from returning to the open position in case of electrical power failure. Upon being energized by the engine failure sensor, these actuators apply a tension force to the long rods running along the front spars. Each of these rods in turn pulls on a load-equalizing pulley cable system at the rotor blade tip. (See Figure 1.) The cable, by pulling on the forward sector and aft lever, rotates both forward and aft valves to the closed position. All driving components are symmetrical; that is, there is an upper and lower cable, sector, connecting rod, and aft lever.

Returning the valves to the open position is accomplished as follows. First, the inboard actuators release their pull; then, if the blades are rotating, gas pressure and centrifugal force are sufficient to force the valves to the open position. In the absence of gas pressure and centrifugal force, the two springs pulling on the section shown in Figure 5 return both valves to their open position.

5. INFLUENCE OF VALVE GEOMETRY AND LOCATION ON ROTOR PERFORMANCE

An aerothermal study to determine the most suitable geometry and location of a valve to provide optimum rotor performance for a one-engine-out emergency condition is summarized in Figure 7. The systems consider partial blockage of each duct at the blade tips (point a); a complete blockage of one duct per blade at the blade roots, with (curve b) and without (curve c) low-pressure duct ventilation to relieve base pressure at the exit of the blocked duct; separate ducts for each engine and also tip blockage of one duct for one engine out and no duct interleakage (point d); a sonic throat at the engine exit to maintain the required engine pressure for on-line operation (point e); and a design which provides no mechanical means to improve rotor power for a one-engine-failed condition (point f).

The gas generator discharge conditions were taken from Reference 3 with the basic twin engine military power operating point corresponding to maximum operating level at the operating limit (maximum allowable turbine inlet temperature and cycle pressure ratio). Gas flow rate and total pressure at the blade tips were found from consideration of tip nozzle flow characteristics, engine operating characteristics, duct losses and centrifugal pumping effects. The losses are related to the duct Mach number and are based on 10 percent total pressure loss for the basic twin engine operating condition. Nozzle thrust efficiency and flow characteristics are taken from Reference 4. Corrections to rotor power are made for those cases in which base drag and/or internal drag are incurred.

Generally, to obtain maximum rotor power, the nozzle exhaust area should be that required to match the exhaust pressure for normal operation of the functioning T64 engine. Also, the used portion of the nozzle exhaust area should be as close to the blade tip as possible to provide the maximum moment arm, and duct pressure and leakage losses should be minimized. Since some of these requirements are mutually exclusive, a compromise is in order.

Utilization of the most outboard position of the cascade nozzle implies complete blockage of the trailing blade ducts since they exhaust inboard of the leading ducts. Blockage may be effected at either blade roots or tips. With the roots blocked, leakage from the forward duct to the rear duct through the flexible joints occurs. A leakage rate of roughly 5 percent is expected and results in a rotor power of 1060 horsepower (point g) compared to the 1380 horsepower available with no leakage (see Figure 7). With the rear duct tips blocked, the leakage

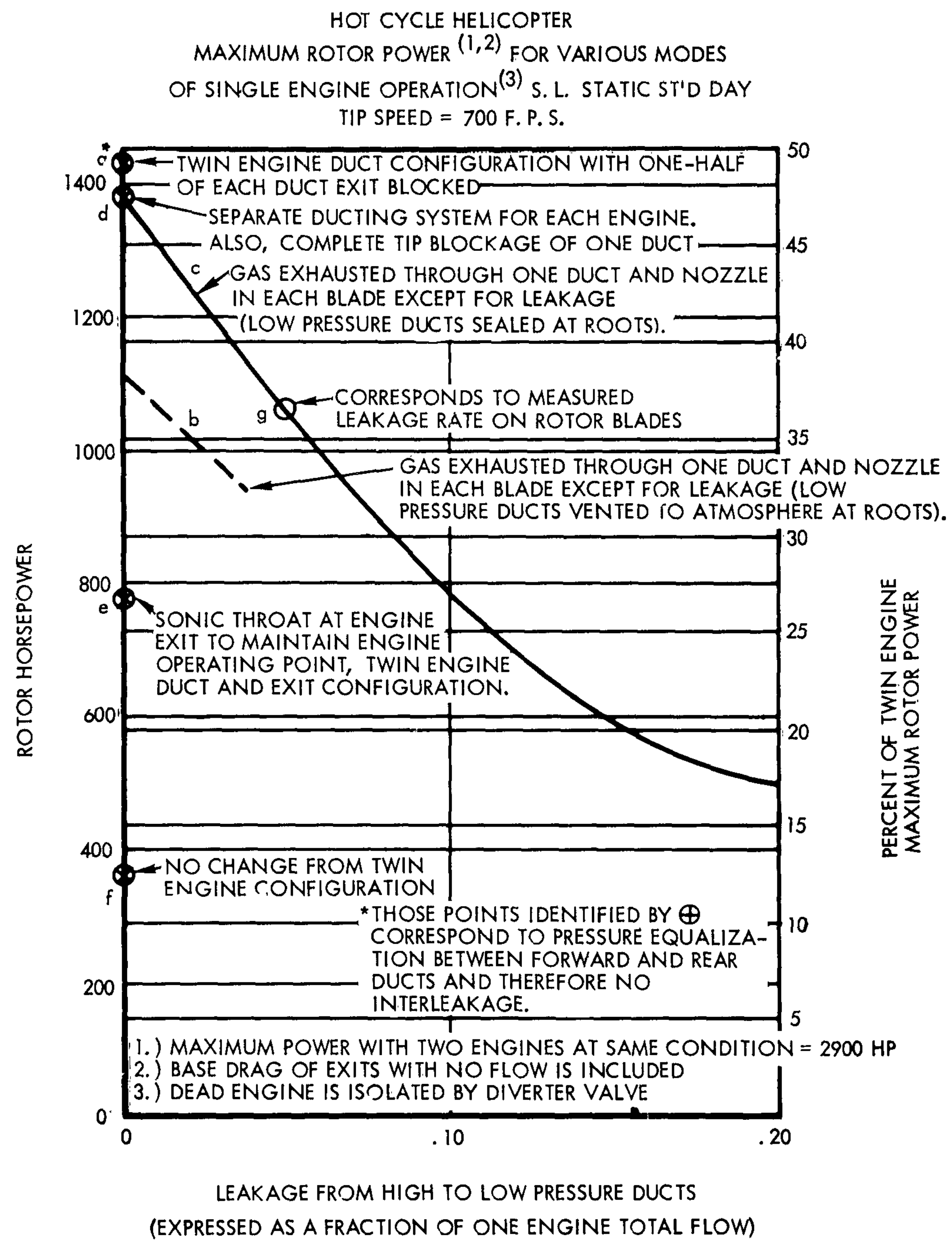


Figure 7. Maximum Rotor Power for Various Modes of Single-Engine Operation.

problem is solved. However, with complete blockage of one duct, all the flow passes through the remaining duct, causing the frictional pressure loss to be four times as great as if the same flow had been ducted through both ducts having half of their exit areas blocked. The latter system provides 1430 horsepower compared to 1380 horsepower for tip blockage (i.e., no leakage) of one duct. Additionally, the partial tip blockage of both ducts eliminates the interleakage problem and, due to reduced vane size, gives the advantage of smaller aerodynamic moments and loads at the valve axle.

The comparison cases for a system using no duct blockage, a sonic throat at engine exit to maintain on-line operation, and low-pressure duct ventilation to relieve base pressure at the exit of a blocked duct all show a drastic reduction in rotor power. Also, it is interesting to note that for a system employing separate ducting for each engine, the thrust corresponds to that of the single duct tip blockage case (1380 horsepower), 50 horsepower less than for the selected system.

The table below provides a comparison of the rotor power yielded by each system and shows the general advantage of the tip valve configuration which provides partial blockage of each duct.

System	Available Rotor hp	Description
1	1430	Partial tip blockage of both ducts
2	1380	Separate duct system for each engine
3	1115	Gas exhausted through one duct, low pressure duct vented, no leakage
4	1060	Gas exhausted through one duct in each blade, 5% leakage
5	780	Sonic throat at engine exit
6	365	No change from twin engine configuration

$$\text{Improvement from original System \#4 to System \#1} = \frac{1430-1060}{1060} = 35\%$$

6. VALVE AERODYNAMIC LOADS AND TEMPERATURE GRADIENTS

Pressure Loads on Vane

The pressure loads on the closed cascade diverting vanes are almost entirely due to duct static pressure since the upstream static pressure is approximately 97 percent of the total pressure. A negative pressure gradient occurs on the vanes in the direction of flow due to the acceleration of fluid in passing from full duct area to the smaller area. This pressure gradient is assumed linear. Also, the rotor tip vortex causes a subambient pressure at the loaded rotor tip, resulting in a downstream pressure on the vanes to be estimated as ($P_{amb} - 1.5 q_{blade}$) at S. L. standard conditions; where q_{blade} is the dynamic pressure at the rotor tip at 240 rpm.

The resulting pressure differentials at the edges of the vanes are given below for two power levels for the T64-GE-6 engine.

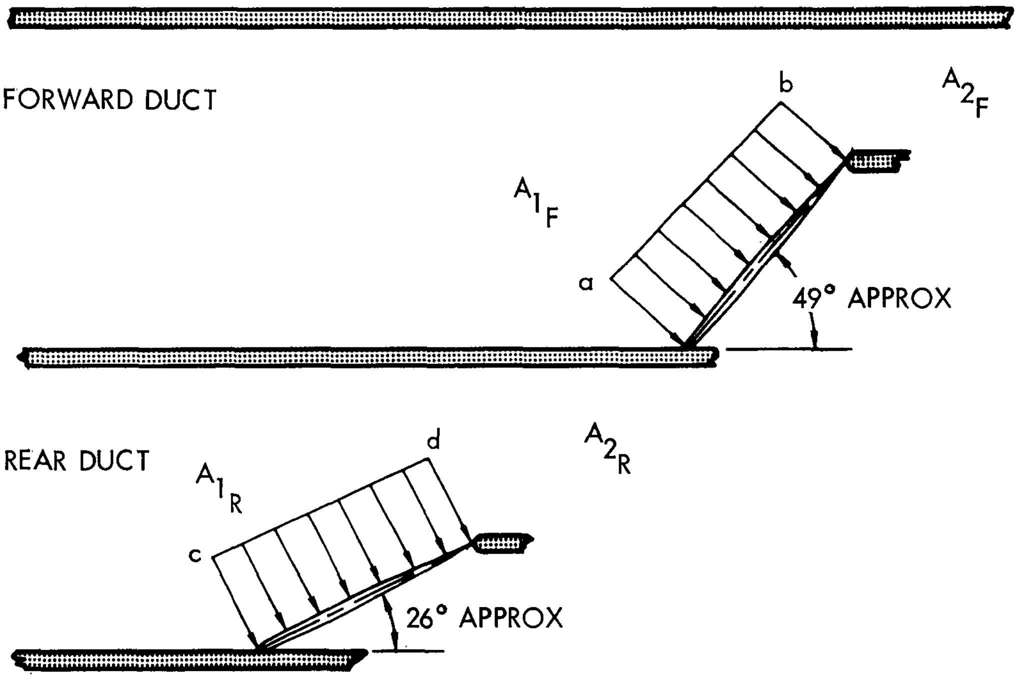
	Max. Power (psi)	Military Power (psi)
Upstream edge of vane in forward duct	32.2	30.8
Downstream edge of vane in forward duct	26.0	25.2
Upstream edge of vane in rear duct	32.2	30.8
Downstream edge of vane in rear duct	31.0	29.6

(Corresponding locations are shown in Figure 8.)

Aerodynamic and Inertia Moments

The forces acting on the cascade diverting vanes were computed to determine if the vanes will remain in the through position until manually closed for one-engine operation. The effects of the weights of the linkage and control system were also considered. Since both vanes are interconnected, the system was analyzed by taking a summation of moments about the forward diverter vane.

PRESSURE DIFFERENTIALS ALONG EDGES OF VANES



$$\frac{A_{2_F}}{A_{1_F}} + \frac{A_{2_R}}{A_{1_R}} = \frac{1}{2.8} + \frac{1}{1.55} = 1.00$$

$$A_{2_F} + A_{2_R} = \frac{1}{2}\left[A_{1_F} + A_{1_R}\right] = A_{1_F} = A_{2_F}$$

Figure 8. Pressure Differentials Along Edges of Vanes.

The conditions investigated are maximum power at 240 rotor rpm and maximum power at zero rotor rpm (conservative estimate simulating starting conditions).

Results indicate that the vane aerodynamic forces create a negative moment (nose down) about the pivot due to the pivot point being located aft of the aerodynamic center. However, as shown in Figure 9, the net resulting moment is positive due to the high moments from the centrifugal force of the mass of the linkage and control system. The large increase in positive moment at approximately 10 degrees vane angle is due to the aft shift of the aerodynamic center to approximately 50 percent chord resulting from vane stall.

The dash curve on Figure 9 shows the conservative estimate of the negative aerodynamic moments about the vane during a starting condition where the assumed gas velocities through the blade ducts are consistent with maximum engine power and zero rotor rpm.

The conditions investigated indicate that the vanes will remain in the through position except during the initial starting conditions.

Listed below are a few of the methods to overcome the negative aerodynamic moments of the vane during starting conditions.

a. Addition of a preload spring to maintain the vanes in the through position (approximately 100 inch-pounds preload).

b. Starting the rotor system on a single engine with vane closed until full rotor rpm is attained.

c. Relocating the vane pivot point forward of the aerodynamic center (1/4 chord).

Method "a" is the system selected for the research vehicle since "b" causes high thermal stresses across the vane and "c" causes a mismatch of present cascade geometry.

Temperature Gradients Across Vane

In order to estimate the temperature drop across the cascade valve, the following assumptions were made:

a. Valve closed after steady state has been reached (transient analysis showed this to be the most serious condition).

b. Temperature on the inboard side of the valve, T_1 = 1200 degrees Fahrenheit.

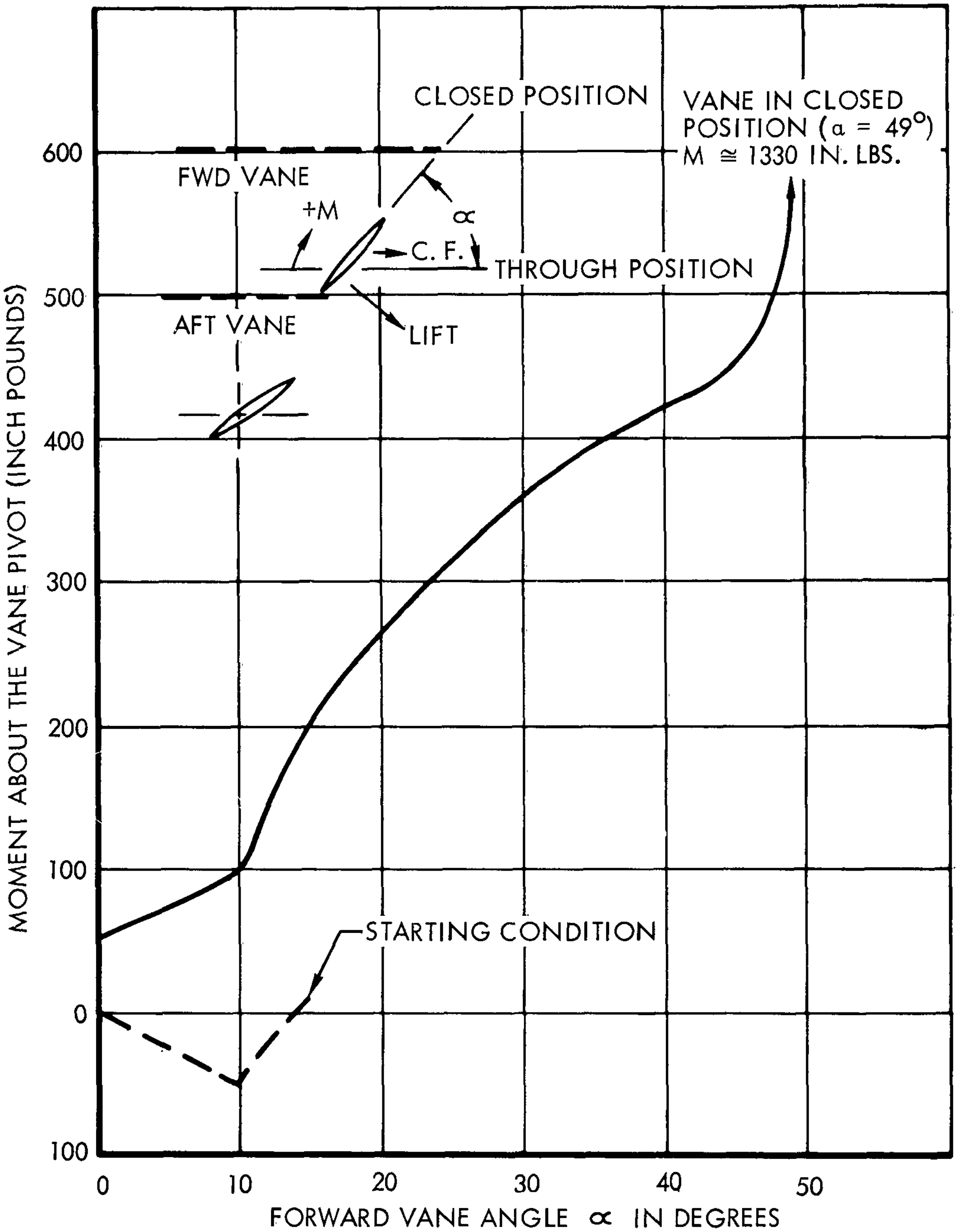


Figure 9. Total Moment About the Forward Vane.

c. Nozzle pressure ratio, NPR = 2.75, γ = 1.35.

d. Emissivity of all metal surfaces are equal, ϵ = 0.8.

e. Heat transmission between the valve and environment is by radiation only.

Extreme cases are analyzed with the environment temperature assumed at different levels to cover probable conditions. Effects of conduction and convection are counteracted by the conduction from hot parts and the leakage of hot gas. The isentropic exhaust conditions corresponding to a nozzle pressure ratio = 2.75 are Mach number = 1.31 and temperature ratio $\left(\frac{T_T}{T}\right)_4$ = 1.3.

Thus, the temperature of the exit jet is T_4 = 1280 degrees R or 820 degrees Fahrenheit.

Nomenclature

T_1 = Temperature of the inboard wall of the valve

T_2 = Temperature of the outboard wall of the valve

T_3 = Temperature of the cascade and vicinity

T_4 = Temperature of the jet

$q/_A$ = Heat transfer quantity per unit area

σ = Stefan Boltzman constant

Case 1. No shielding from exhaust jet by cascade.

T_1 = 1200°F

T_3 = 820°F; equal to the temperature of the jet

$T_3 = T_4$

$$\frac{q}{A_2} = T_1^4 - T_2^4 = T_2^4 - T_3^4$$

$$T_2^4 = \frac{T_1^4 + T_3^4}{2} = \frac{7.57 + 2.69}{2} \text{ X } 10^{12} = 5.13 \text{ X } 10^{12}$$

$$T_2 = 1510°R \text{ or } 1050°F$$

$$\Delta T = T_1 - T_2 = 1200 - 1050 = 150°F$$

Case 2. Complete shielding from exhaust jet by cascade which is assumed to be at 600°F.

$$T_1 = 1200°F$$

$$T_3 = 600°F$$

$$T_2^4 = \frac{7.57 + 1.27}{2} 10^{12}; \quad T_2 = 1450°R \text{ or } 990°F$$

$$\Delta T = T_1 - T_2 = 1200 - 990 = 210°F$$

Case 3. Complete shielding by cascade assumed to be at 400°F.

$$T_1 = 1200°F$$

$$T_3 = 400°F \text{ or } 860°R$$

$$T_2^4 = \frac{7.57 - 0.55}{2} \times 10^{12} = 1.37 \times 10^{12}$$

$$T_2 = 1370°R \text{ or } 910°F$$

$$\Delta T = T_1 - T_2 = 1200 - 910 = 290°F$$

Case 4. Complete shielding by cascade which exchanges radiation with the exhaust gas. (For this case, radiation of the hot gas is taken into account). T_2 and T_3 are unknown.

$$T_1 = 1200°F$$

$$T_4 = 820°F$$

It is also assumed that the cascade (represented by T_3) on the outside exchanges radiation with the hot gas only.

$$\frac{q}{A\sigma} = 0.8(T_1^4 - T_2^4) = 0.8(T_2^4 - T_3^4) \text{ then } T_3^4 = 2T_2^4 - T_1^4$$

also (see Reference 5 for definition of constants)

$$\frac{q}{A\sigma} = \epsilon_S' \left[{}_GT_4^{\ 4} - \alpha_G T_S^{\ 4} \right] = \frac{0.8 + 1}{2} \left[0.1\ T_4^{\ 4} - 0.9\ T_3^{\ 4} \right]$$

$$= 0.9 \left[0.1T_4^{\ 4} - 0.9T_3^{\ 4} \right] = 0.9 \left[0.1T_4^{\ 4} - 0.9\ (2T_2^{\ 4} - T_1^{\ 4}) \right]$$

$$T_2 = 1490^\circ R = 1030^\circ F$$

$$\Delta T = T_1 - T_2 = 1200 - 1030 = 170^\circ F$$

On the basis of calculations of Case 1 through Case 4 with assumptions as stated, the temperature difference between the valve walls between the ribs does not exceed 300 degrees Fahrenheit. The temperature difference on the ribs of the valve is estimated not to exceed 200 degrees Fahrenheit.

7. REFERENCES

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4. Jones, D. L. and Rabek, J. W.; "Results of Static Test Program - Gas Flows and Temperature," HTC-AD Report No. 285-9-7, February 1962.

5. McAdams, W. H.; Heat Transmission, 3rd Edition, McGraw Hill, Inc., New York, N. Y., 1964, page 68.

6. MIL-HDBK-5, Strength of Metal Aircraft Elements, March 1959.

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10. Report 285-13 (62-13) Contract No. AF 33(600)-30271 "Hot Cycle Rotor System Structural Analysis, Vol. I," March 1962.

11. Report 285-13(62-13) Contract No. AF33(600)-30271, "Hot Cycle Rotor System Structural Analysis, Vol. II," March 1962.

12. Memo from R. T. Neher to E. Sallows, Hughes Tool Company - Aircraft Division, dated 16 April 1962, "Pressure Loads on Cascade Inlet Diverting Vanes During One Engine Operation."

13. Memo from C. R. Smith to J. L. Velazquez, Hughes Tool Company - Aircraft Division, dated 25 February 1962, "Pressure Limitation for Hot Cycle Blade Ducts."

APPENDIX A

STRESS ANALYSIS

Duct Closure Valves

The following pages contain the stress analysis of the cascade closure valves and the operating mechanism. Loads on the valves are due to

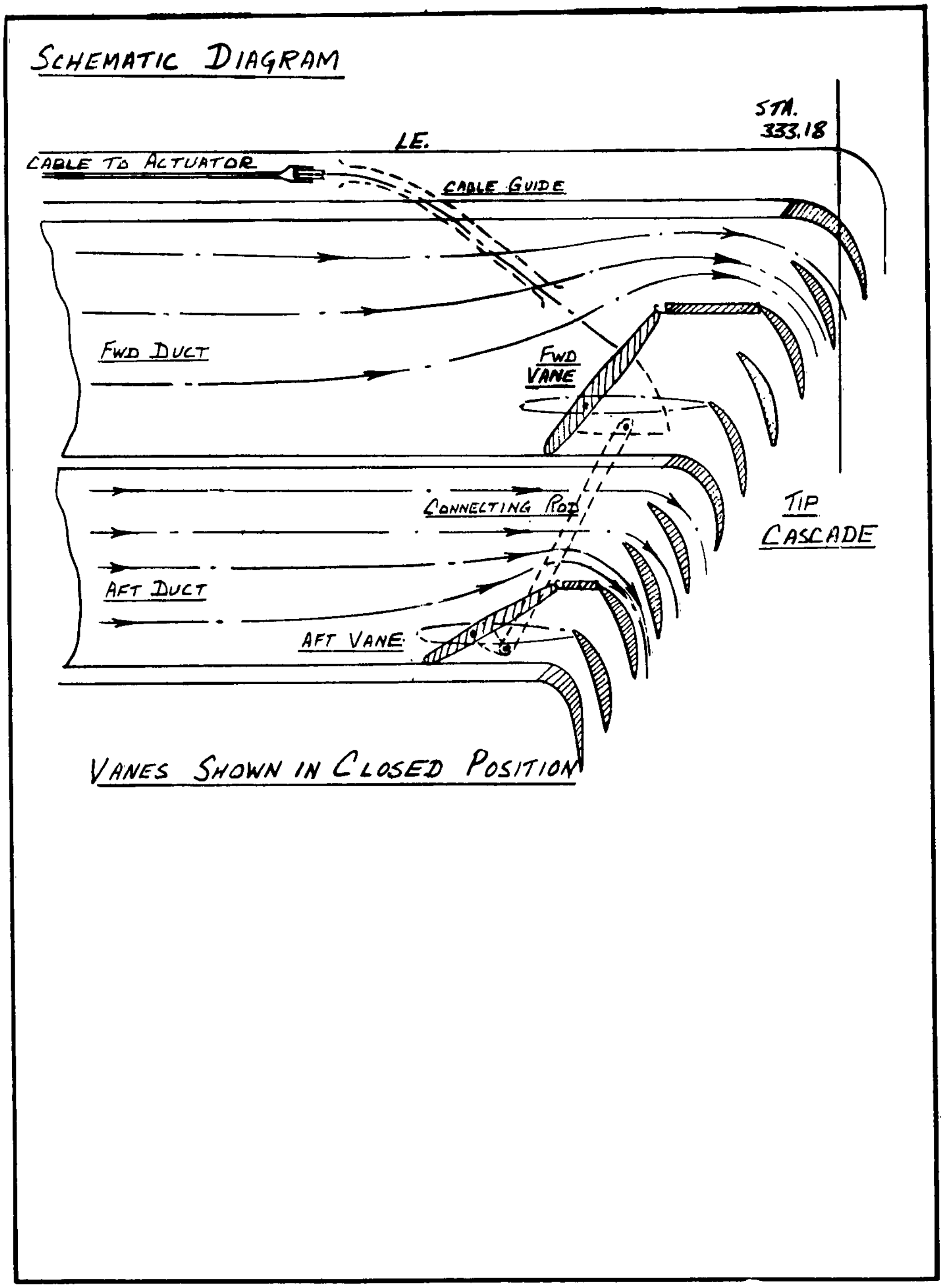
i) Pressure
ii) Centrifugal effects
iii) Thermal gradients.

Information on (i) was obtained from Reference 12.
Information on (ii) was obtained from Reference 10.
Information on (iii) was obtained from Pages 33-35.

Analysis of the basic tip structure was made where this varied from the previous configuration, particularly in the areas where the valves are mounted. However, a general analysis of the structure inboard of the new tip assembly (which will have increased centrifugal loadings, due to increase of tip weight) has not been performed at this date and will be part of the redesign to steel spars.

Concept of Single Valve Located Inboard

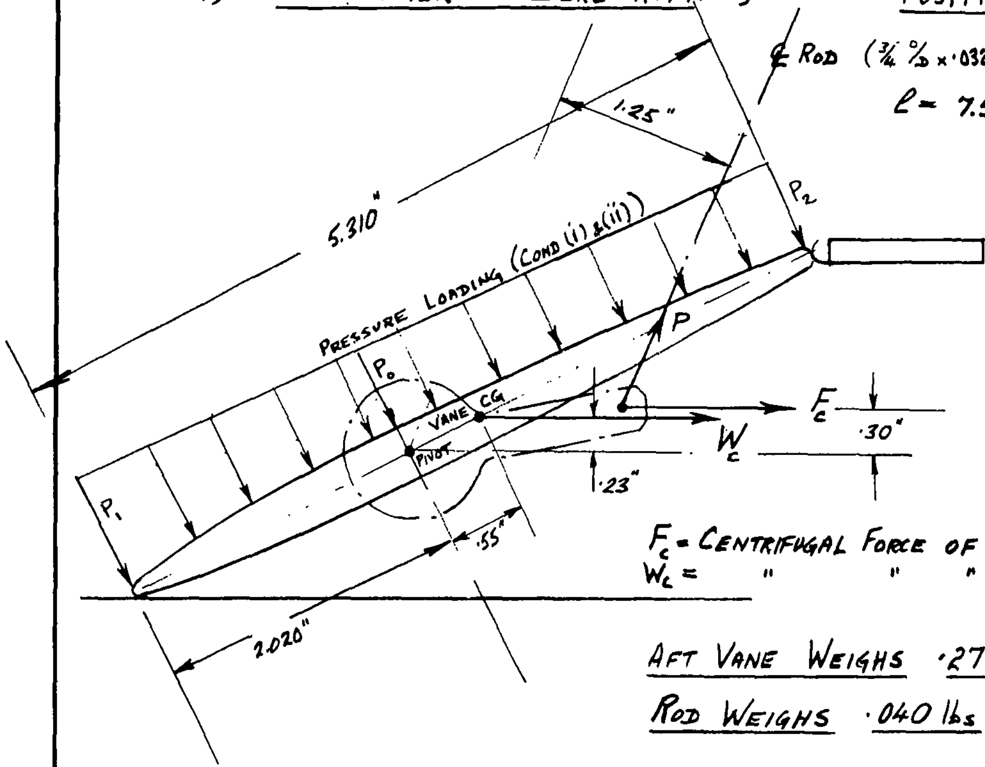
The analysis of one duct operating alone, based on the concept of a valve located inboard, is found on page 80. This analysis indicates that the basic support structure for the duct is understrength for such a configuration.


SCHEMATIC DIAGRAM
STA.
333.18
L.E.
CABLE TO ACTUATOR
CABLE GUIDE
FWD DUCT
FWD
VANE
CONNECTING ROD
TIP
CASCADE
AFT DUCT
AFT VANE
VANES SHOWN IN CLOSED POSITION

FORCES ON AFT VANE

TWO CONDITIONS CONSIDERED

i) MAX. POWER + OVER REV.
ii) MAX POWER + ZERO RPM.
} VANE IN CLOSED POSITION.



F_c = CENTRIFUGAL FORCE OF ROD
W_c = " " " VANE

AFT VANE WEIGHS .27 lbs
ROD WEIGHS .040 lbs

PRESSURE LOADINGS
REF # 12

P_1 = 32.2 psi (LIMIT) × 2.0 = 64.4 #/in² (ULT.)
P_2 = 31.0 psi. (LIMIT) × 2.0 = 62.0 #/in² (ULT.)

CENTRIFUGAL LOADING :— OVER REV (875 f.p.s) (REF. # 5 p.4.2.2)

AT STA. 333" "g" FACTOR = 857

$F_c = 857 \times \frac{.04}{2}$ = 17 lb. (LIMIT.) × 1.5 = 25.5 lbs (Ult) (PER ROD)
$W_c = 857 \times .27$ = 231 lb. (LIMIT) × 1.5 = 347 lbs (Ult) (TOTAL)

Aft Vane (Cont'd)

Cond (i) Max Power + 240 RPM.

Pressure Load at Pivot

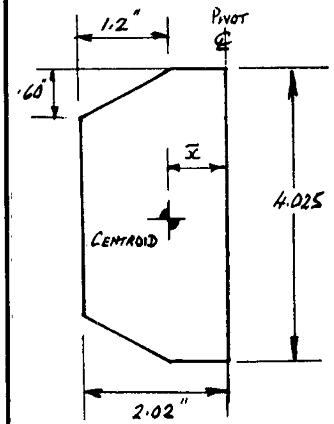
$$P_o = 62.0 + \frac{(64.4 - 62.0)}{5.31} \times (5.31 - 2.02)$$

$$= \underline{\underline{63.49 \text{ lbs/in}^2}}$$

Moments About Pivot ($M_o = \circlearrowright$ +ve)

$$\frac{64.4 + 63.49}{2} = \underline{\underline{63.94 \text{ lbs/in}^2}} \quad \text{(Avge. Pressure fwd. of Pivot)}$$

Area Fwd of Pivot



$$A = 4.025 \times 2.02 = 8.14$$
$$- 1.20 \times .60 = .72$$
$$\underline{7.42 \text{ in}^2}$$

$$A\bar{x} = 8.14 \times 1.01 = 8.230$$
$$- .72 \times 1.62 = 1.165$$
$$\underline{7.065 \text{ in}^3}$$

$$\bar{x} = \frac{7.065}{7.42} = .952''$$

$$M_o = \bar{7}.065 \times 63.94 = \underline{\underline{\overline{452} \text{ lbs ins}}}$$

Area Aft of Pivot. $3.29 \times 4.025 = \underline{\underline{13.25 \text{ in}^2}}$

$$\bar{x} = 1.645''$$
$$A\bar{x} = 21.80 \text{ in}^3$$

$$P_{AVGE} = \frac{63.49 + 62.00}{2} = \underline{\underline{61.745 \text{ lbs/in}^2}}$$

$$M_o = 21.80 \times 61.745$$
$$\doteq \underline{\underline{1345 \text{ lbs. ins}}}$$

$F_c = 25.5$ lbs $\quad M_o = 25.5 \times .30 \times 2 = \underline{15 \text{ lbs ins}}$ (Two Rods)

$W_c = 3.47$ lbs $\quad M_o = 3.47 \times .23 = \underline{80 \text{ lbs ins}}$

Totals.

Cond (i) $\quad 1345 - 452 + 15 + 80 = \underline{\underline{988 \text{ lbs ins.}}}$ $\quad \underline{\underline{P = 791 \text{ lbs}}}$

Cond (ii) $\quad 1345 - 452 = \underline{\underline{893 \text{ lbs. ins}}}$ $\quad \underline{\underline{P = 715 \text{ lbs}}}$

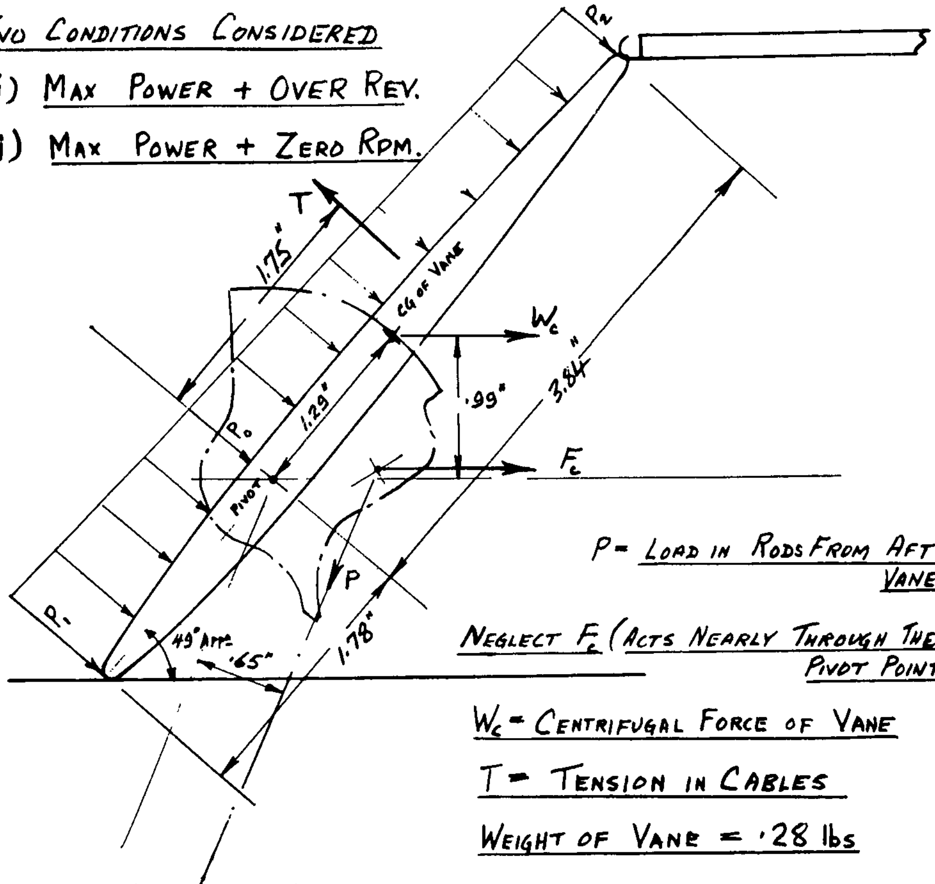
P = Load in Two Rods

FORCES ON FWD VANE (VANE IN CLOSED POSITION)

TWO CONDITIONS CONSIDERED

(i) MAX POWER + OVER REV.

(ii) MAX POWER + ZERO RPM.



P = LOAD IN RODS FROM AFT VANE

NEGLECT F_c (ACTS NEARLY THROUGH THE PIVOT POINT)

W_c = CENTRIFUGAL FORCE OF VANE

T = TENSION IN CABLES

WEIGHT OF VANE = .28 lbs

PRESSURE LOADS ON VANE (REF # 12)

$P_1 = 32.2\ psi.\ (LIMIT) \times 2.0 = 64.4\ \#/in^2\ (Ult.)$

$P_2 = 26.0\ psi.\ (LIMIT) \times 2.0 = 52.0\ \#/in^2\ (Ult.)$

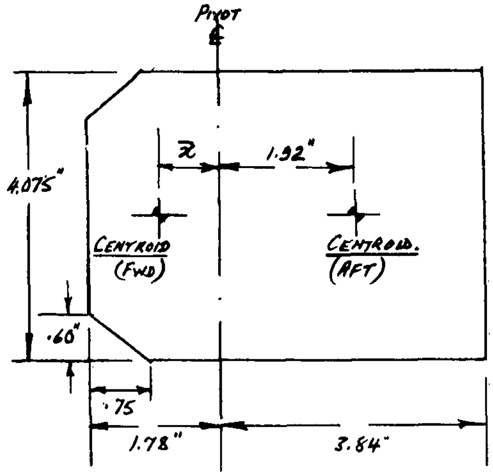
$$P_o\ (\text{AT PIVOT}) = 52.0 + \frac{(64.4-52)}{5.62} \times 3.84 = 52 + 8.45 = 60.45\ \#/in^2\ (Ult.)$$

AVGE. PRESSURE ON NOSE PORTION $= \frac{64.4 + 60.45}{2} = 62.42\ \#/in^2$

AVGE PRESSURE ON AFT PORTION $= \frac{52.0 + 60.45}{2} = 56.22\ \#/in^2$

CENTRE OF LOAD LINE FOR AFT PORTION ASSUMED TO LIE ON CENTROID OF AFT PORTION (CONSERVATIVE)

FWD. VANE (CONT^D)



NOSE PORTION

$$A = 4.075 \times 1.78 = 7.26$$
$$- .60 \times .75 = \underline{.45}$$
$$6.81\ in^2$$

$$A\bar{x} = 7.26 \times .89 = 6.46$$
$$- .45 \times 1.53 = \underline{.69}$$
$$5.77$$

$$\bar{x} = .85''$$

$$M_0 = \bar{5}.77 \times 62.42 = \underline{-360\ lbs\ ins.}$$

AFT PORTION

$$A = 3.84 \times 4.075 = 15.65\ in^2$$
$$A\bar{x} = 15.65 \times 1.92 = 30.00\ in^3$$

$$M_0 = 30 \times 56.22 = \underline{1685\ lbs\ ins}$$

$$\text{NETT PRESSURE } M_0 = 1685 - 360 = \underline{1325\ lbs\ ins.}$$

$$W_L = 857 \times .28 = 240\ lbs\ (LIMIT)$$
$$= \underline{360\ lbs\ (ULT)} \qquad M_0 = 360 \times .99 = \underline{356\ lbs.ins}$$

$$P = 791\ (COND\ i)\ \&\ 715\ (COND\ ii) \qquad M_0 = 791 \times .65 = 514\ lbs.ins$$
$$= 715 \times .65 = 464\ lbs.ins.$$

TOTALS.

$$COND(i) \quad M_0 = 1325 + 356 + 514 = \underline{2195\ lbs.\ ins}$$
$$COND(ii) \quad M_0 = 1325 + 464 = \underline{1789\ lbs.\ ins}$$

TENSION IN CABLES. (T = LOAD IN TWO CABLES)

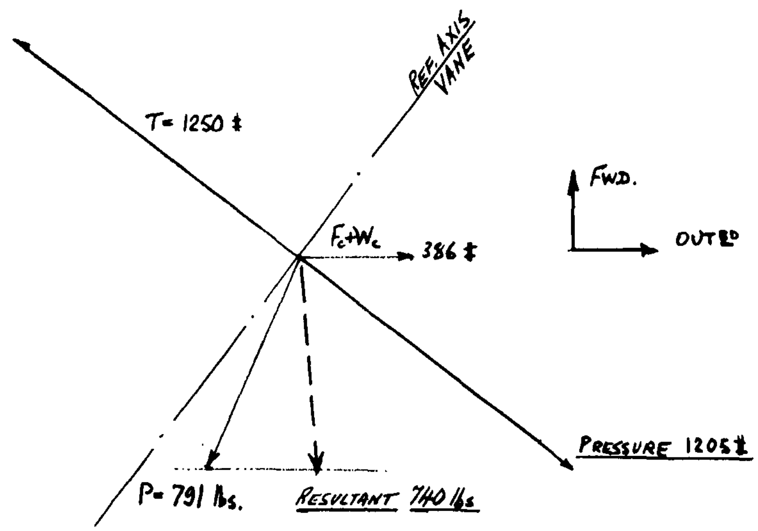
$$\underline{COND\ (i)} \quad T = 1250\ lbs\ (ULT.)$$
$$\underline{COND\ (ii)} \quad T = 1023\ lbs\ (ULT.)$$

$$\text{LIMIT LOAD (FOR ACTUATOR)} = \frac{\left\{\frac{988}{2} + \frac{95}{1.5}\right\} \times \frac{.65}{1.23} + \frac{1325}{2} + \frac{356}{1.5}}{1.75} = \underline{680\ lbs}$$

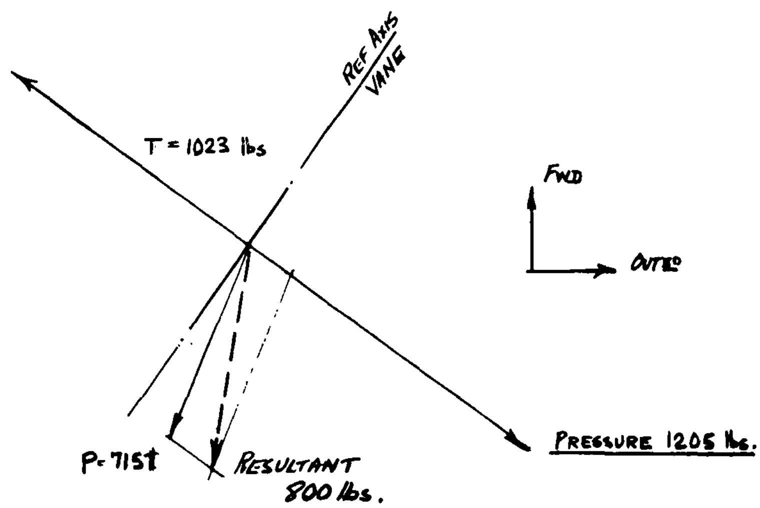
LOADS ON PIVOT POINTS

a) FORWARD VANE NOTE:- THESE ARE TOTAL LOADS TO PIVOT (NOT SPLIT INTO TOP & BOTTOM REACTIONS)

COND (i)



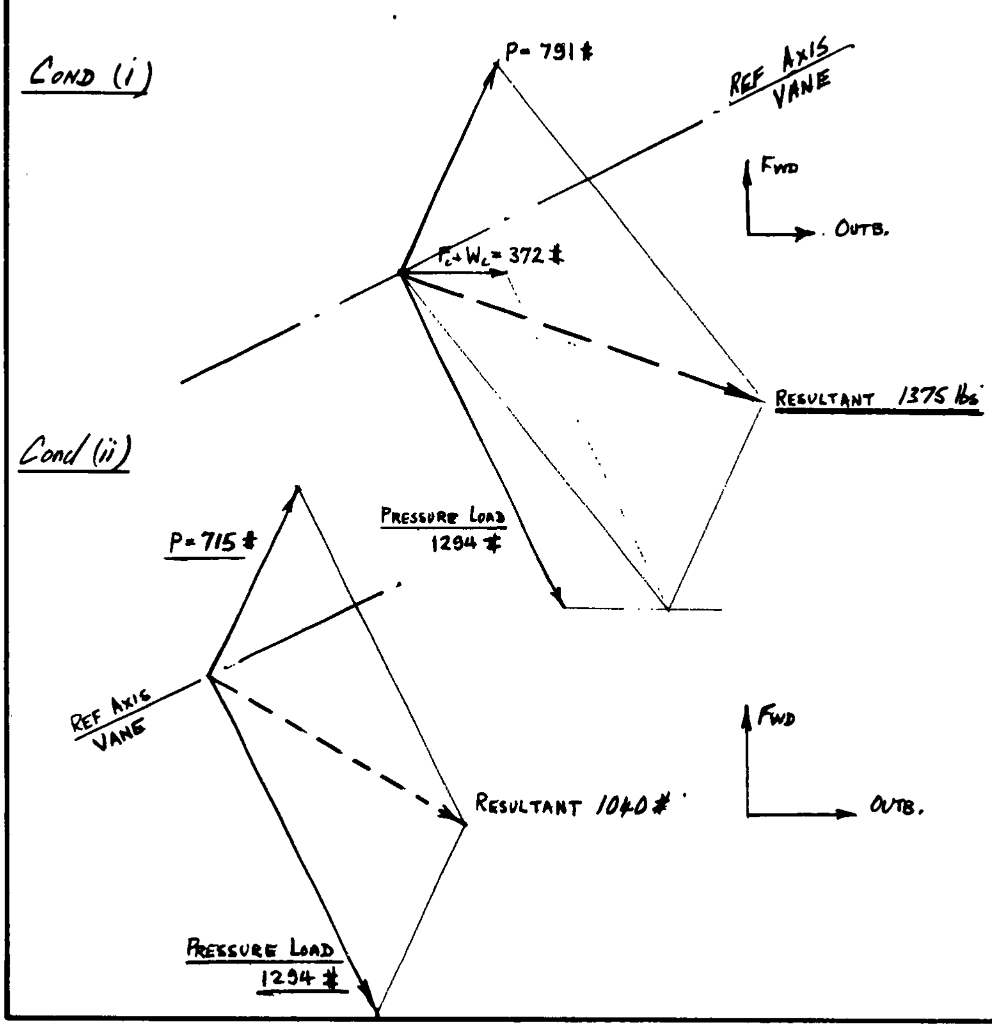
COND (ii)



LOADS ON PIVOT POINTS

b) AFT VANE

NOTE:- THESE ARE TOTAL PIVOT LOADS (NOT SPLIT IN TOP & BOTTOM REACTIONS)



STRESS ANALYSIS OF VANES

FORWARD VANE ——— DWG. Nº 385-1104

AFT. VANE ——— DWG Nº 385-1107

FROM AN EXAMINATION OF SIZE, OVERHANG FROM PIVOT, AND GENERAL LOADING, IT CAN BE SEEN THAT THE FORWARD VANE IS THE MORE CRITICAL OF THE TWO. THE ANALYSIS OF THE FORWARD VANE COVERS THE AFT VANE, AS THE SECTIONS OF RIBS, SPARS ETC WERE MADE SIMILAR.

SINCE THE ANALYSIS OF THE FWD. VANE WAS COMPLETED, AN INCREASE IN VANE THICKNESS HAS TAKEN PLACE. THIS MEANS THAT THE TORSION BOX SIZES HAVE INCREASED AND THE SHEARS DETERMINED ON PAGE WILL BE DECREASED. THIS CHANGE HAS NOT BEEN CORRECTED AND SO THE TORSION SHEARS ARE CONSERVATIVE. HOWEVER, THE INCREASED SIZE HAS BEEN USED FOR CHECKING OF RIB SECTIONS FOR BENDING (ETC).

THE MATERIAL OF THE VANES IS RENÉ 41, THE SKINS BEING BRAZED TO THE MACHINED FRAMEWORK. INFORMATION ON THIS BRAZING WAS TAKEN FROM THE CATOLOGUE REF. # 9 THIS GIVES THE SHEAR STRENGTH OF A LAP JOINT AS 30,000 psi. AT 1200°F. BECAUSE OF — a) THE LACK OF INFORMATION ON CREEP-RUPTURE STRENGTH OF THIS BRAZE

b) IMPROBABILITY OF OBTAINING 100% SURFACE BONDING WITH SMALL OVERLAP DISTANCES

c) CATOLOGUE ALLOWABLES ARE FOR THE BRAZING STAINLESS STEELS. WE ARE USING RENE 41 WHICH MAY LOWER THE BRAZE ALLOY STRENGTH,

— THE ALLOWABLE BRAZE SHEAR STRENGTH WAS TAKEN TO BE 10,000 psi. THIS IS THOUGHT TO BE CONSERVATIVE.

FORWARD VANE

THERMAL STRESSES

a) TEMP ON INSIDE = 1200°F

b) TEMP GRADIENT ACROSS VANE = 200°F

SINCE THE STRUCTURE IS NOT RESTRAINED (IE. IT IS FREE TO DEFLECT IN BENDING) AND THE TEMPERATURE GRADIENT IS LINEAR, THE THERMAL STRESSES ARE THEORETICALLY ZERO

HOWEVER, THERE IS A SMALL GRADIENT BETWEEN THE CENTER OF THE PANELS AND THE EDGES, ON THE HOT SIDE OF THE VANES.

ASSUME i) FULLY RESTRAINED PANEL
ii) GRADIENT = 50°F

$$\alpha = 8 \times 10^{-6} \text{ in/in per } {}^\circ F$$

$$\delta = 8 \times 50 \times 10^{-6} \times \ell$$

$$\frac{f_c \ell}{E} = \delta$$

$$\therefore \quad f_c = E \times 8 \times 50 \times 10^{-6}$$

$$E = 24 \times 10^{6}$$

$$f_c = 24 \times 400$$

$$= \underline{9600 \text{ lbs/in}^2}$$

THERMAL STRESSES WOULD RELIEVE THE STRESSES DUE TO PRESSURE LOADS.

AS THE THERMAL STRESSES ARE LOW, THIS RELIEF WILL BE NEGLECTED

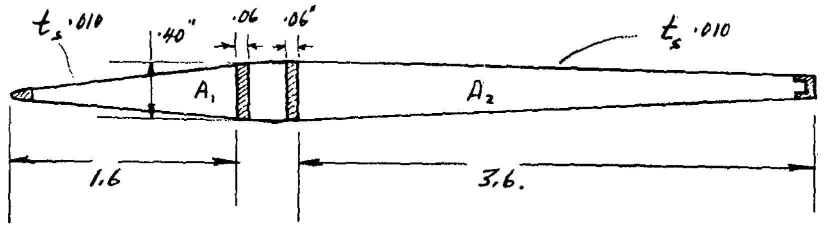
FORWARD VANE (CONT'D)

$$\text{EULER } P_{CR} = \frac{2\pi^2}{1.3} = 15.2 \text{ lbs per } 1'' \text{ strip}$$

$A = .010$ $\quad \therefore f_{cr} = \dfrac{15.2}{.010} = 15{,}200$ psi.

THIS INDICATES THAT THE SKIN DOESN'T BUCKLE UNDER THE THERMAL STRESSES. (IF FULLY RESTRAINED AT EDGES)

SECTION OF VANE



A_1 = AREA — NOSE TORSION BOX — $1.6 \times \dfrac{.4}{2} = .32 \text{ in}^2$

A_2 = AREA — AFT TORSION BOX — $3.6 \times \dfrac{.4}{2} = .72 \text{ in}^2$

RELATIVE STIFFNESS DETERMINES DISTRIBUTION OF TORQUE

TOTAL TORQUE = T_1 $\qquad$ NOSEBOX TORQUE = T_1

$$\frac{T_1}{T_0} = \frac{a_{12}\mu_1 + a_2}{a_1\gamma_1^2 + a_2 + a_{12}\mu_1^2}$$

$$a_{12} = \frac{p_{12}}{G_{12} t_{12}} = \frac{.4}{G_{12} \cdot .12} \qquad a_1 = \frac{p_1}{G_1 t_1} = \frac{3.2}{G_1 \cdot .01}$$

$$a_2 = \frac{p_2}{G_2 t_2} = \frac{7.2}{G_2 \cdot .01} \qquad G = \text{CONSTANT}$$

$$a_{12} = 3.33 \qquad a_1 = 320 \qquad a_2 = 720$$

FWD VANE (CONT'D)

$$\gamma_1 = \frac{A_2}{A_1} = \frac{.72}{.32} = 2.25 \qquad \gamma_1^2 = 5.05$$

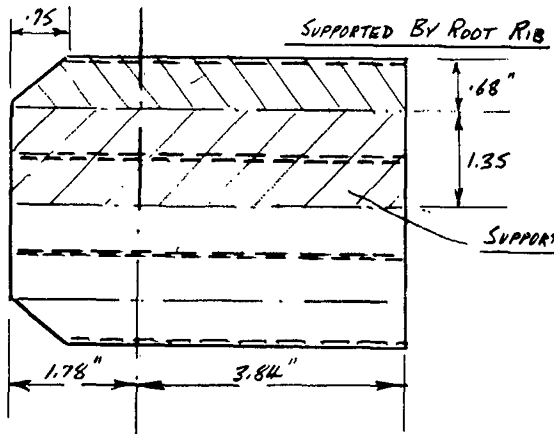
$$\mu_1 = 1 + \frac{A_2}{A_1} = 3.25 \qquad \mu_1^2 = 10.6$$

$$\frac{T_1}{T_0} = \frac{3.33 \times 3.25 + 720}{320 \times 5.05 + 720 + 3.33 \times 10.6} = .272$$

$$T_1 = .272\,T_0 \qquad q_1 = \frac{T_1}{2A_1} = \frac{.272\,T_0}{.64} = \underline{\underline{.435\,T_0}}$$

$$T_2 = .728\,T_0 \qquad q_2 = \frac{T_2}{2A_2} = \frac{.728\,T_0}{1.44} = \underline{\underline{.505\,T_0}}$$

PRESSURE LOADING ON RIBS.



THE CENTRIFUGAL LOADING WILL BE DISTRIBUTED IN A SIMILAR WAY.

TOTAL $W_c = 360$ lbs. RESOLVE @ $49° = W_c \sin 49°$
$= 272$ lbs

ROOT RIB $= \frac{272}{6} = 45.3$ lbs $\quad w = \frac{45.3}{4.87} = \underline{\underline{9.3\ \#/in}}$

INT. RIB $= \frac{272}{3} = 91$ lbs. $\quad w = \frac{91}{5.62} = \underline{\underline{16.2\ \#/in}}$.

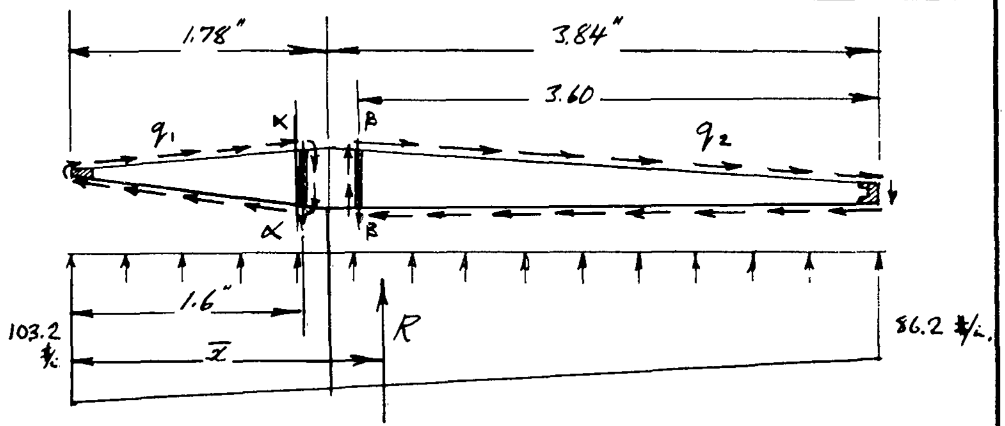
THESE VALUES WILL BE ADDED TO PRESSURE LOADING

FORWARD VANE (CONT'D)

INTERMEDIATE RIBS

L.E. LOAD $= 1.35 \times 64.4 = 87$ lbs/in $+ 16.2 = \underline{\underline{103.2 \text{ lbs/in.}}}$

T.E. LOAD $= 1.35 \times 52.0 = 70.2$ lbs/in $+ 16.2 = \underline{\underline{86.2 \text{ lbs/in}}}$



$$R = 86.2\{1.78 + 3.84\} + \frac{(103.2 - 86.2)(3.84 + 1.78)}{2}$$

$$= 486 + 48 = \underline{\underline{534 \text{ lbs}}}$$

$$R\bar{x} = 486 \times 2.81 + 48 \times 1.873 = 1367 + 90 = \underline{\underline{1457 \text{ lbs ins}}}$$

$$\bar{x} = 2.73"$$

$$T_0 = (2.73 - 1.78) \times 534 = \underline{506 \text{ lbs ins.}}$$

$$q_1 = .435\, T_0 = \underline{220 \text{ lbs/in.}}$$

$$q_2 = .505\, T_0 = \underline{256 \text{ lbs/in.}}$$

DIRECT SHEAR LOAD (ON SPAR) $= \underline{534 \text{ lbs}}$.

$.5"$ DEEP $\therefore q_s = \frac{534}{.50} = \underline{\underline{1068 \text{ lbs/in.}}}$

KK RIB SHEAR $= \frac{103.2 + 98.4}{2} \times 1.6 + .4 \times 220$

$$= 162 + 88 = \underline{\underline{250 \text{ lbs}}}$$

FWD VANE (CONT'D.) INTERMEDIATE RIBS

RIB SHEAR, $\beta\beta$ (3.6 FWD OF T.E.)

$$V_{\beta\beta} = \frac{98.4 + 86.2}{2} \times 3.6 - .4 \times 256$$

$$= 332 - 102 = \underline{230 \text{ lbs}}$$

BENDING MOMENTS

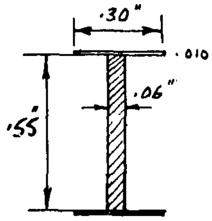
$$M_{\alpha\alpha} = 2A_1 q_1 + 162 \times .9 \text{ (Approx)}$$

$$= 2 \times .32 \times 220 + 146 = \underline{287 \text{ lbs. ins.}}$$

$$M_{\beta\beta} = 2A_2 q_2 - 332 \times 1.8 \text{ (Approx)}$$

$$= 2 \times .72 \times 256 - 597 = \underline{228 \text{ lbs. ins}}$$

SECTION $\alpha\alpha$



$$I_{NA} = .003 \times 2 \times .275^2 + \frac{.55^3 \times .06}{12}$$

$$= .000454 + .000839$$

$$= \underline{\underline{.001293 \text{ in}^4}}$$

$$f = \frac{Mc}{I} = \frac{287 \times .275}{.001293} = \underline{61{,}000 \text{ lbs/in}^2}$$

M.S. HIGH

BRAZE SHEAR $q = \frac{VQ}{I}$ $Q = .003 \times .275 = .000825$

$$q = \frac{250 \times .000825}{.001293} = 159 \text{ lbs/in}$$

+ TORQUE SHEAR 220 lbs/in

$$\text{TOTAL} = 220 + 159 = \underline{379 \text{ lbs/in.}}$$

$$f_s = \frac{379}{.06} = \underline{6320 \text{ psi}}$$

Allowable 10,000 psi. M.S. +.58

NOTE

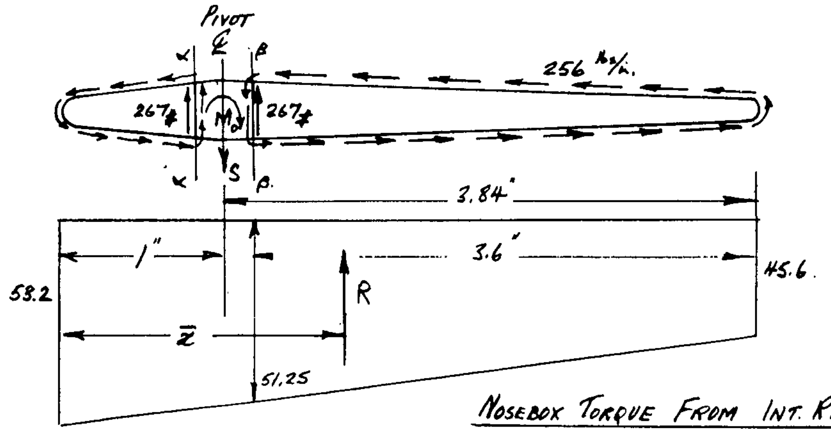
BRAZE ALLOWABLE IS CONSERVATIVELY ESTIMATED FROM VENDOR DATA. (SEE REFERENCE # 9.)

FORWARD VANES (CONT'D)

END RIBS

LE PRESSURE LOAD = 64.4 x .68 = 43.9 + 9.3 = 53.2 #/in

T.E PRESSURE LOAD = 52.0 x .68 = 35.3 + 9.3 = 45.6 #/in



NOSEBOX TORQUE FROM INT. RIB = 141 lbs in

NEW TORQUE BOX AREA = .254 in²

$$q_1 = \frac{141}{2 \times .254} = 278 \text{ lbs/in.}$$

$$R = 45.6 \times 4.84 + \frac{7.6 \times 4.84}{2} = 221 + 18 = 239 \text{ lbs.}$$

$$R\bar{x} = 221 \times 2.42 + 18 \times 1.61 = 535 + 29 = 564 \text{ lbs ins}$$

$$\bar{x} = 2.36$$

$$M_o = 506 + 239 \times 1.36 = 325 + 506 = 831 \text{ lbs.ins}$$

$$S = 267 + 267 + 239 = 773 \text{ lbs.}$$

SHEAR AT ββ

$$V = \frac{51.25 + 45.6}{2} \times 3.6 + .4 \times 256$$

$$= 174 + 102 = 276 \text{ lbs.}$$

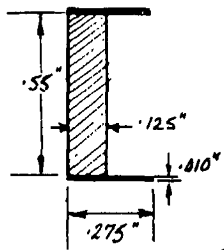
BENDING MOMENTS

$$M_{\beta\beta} = 2A_2 q_2 + 174 \times 1.8 \text{ (Appx)}$$

$$= 2 \times .72 \times 256 + 313 = 682 \text{ lbs ins}$$

FORWARD VANE (CONT'D.) END RIBS

SECTION BB.



$$I_{NA} = 2 \times \cdot 00275 \times \cdot 275^2 + \frac{\cdot 125 \times \cdot 55^3}{12}$$

$$= \cdot 000416 + \cdot 00174$$

$$= \cdot 002156 \text{ in}^4$$

$$f = \frac{Mc}{I} = \frac{682 \times \cdot 275}{\cdot 002156} = 87{,}000 \text{ lbs/in}^2.$$

BRAZE SHEAR $q = \frac{VQ}{I}$ $Q = \cdot 00275 \times \cdot 275$
$= \underline{\underline{\cdot 00076 \text{ in}^3}}$

$$q = \frac{276 \times \cdot 00076}{\cdot 002156} = 97 \text{ lbs/in}.$$

+ TORQUE SHEAR

TOTAL - $97 + 256 = \underline{\underline{353 \text{ lbs/in.}}}$

TO THIS MUST BE ADDED THE MEMBRANE PULL (PAGE)
+ T.F.

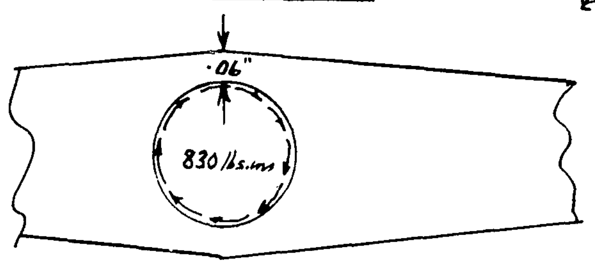
$w = 421$ lbs/in $M = 71$ lbs ins

$Z = \frac{\cdot 55 \times \cdot 125^2}{6} = \cdot 00153$ $f = 46{,}500$ lbs/in^2
+ 87,000

TOTAL BRAZE SHEAR $\sqrt{421^2 + 353^2} = 550$ #/in. Total = 133,500 lbs/in^2

$f_s = \underline{\underline{4400 \text{ lbs/in}^2}}$. @ 154,000 M.S +.14

SECTION ACROSS SPLINES



$$\text{NETT } M_{\mathfrak{C}} = 2A_2q_2 + 186 \times 1.9 + 267 \times \cdot 24 - \frac{830}{2}$$

$$= 369 + 353 + 64 - 415$$

$$= \underline{371 \text{ lbs ins}}$$

$$\frac{M}{d} = \frac{371}{\cdot 50} = 742 \text{ lbs}$$

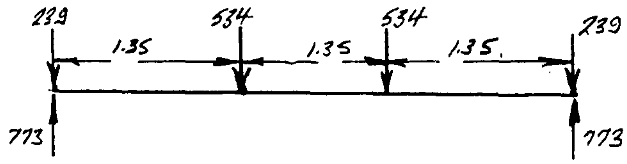
$$A = \cdot 06 \times \cdot 12 = \cdot 0072$$

$$\frac{P}{A} = \underline{\underline{103{,}000 \text{ lbs/in}^2}}$$

@ 150,000 M.S. +.45

FORWARD VANE (CONT'D)

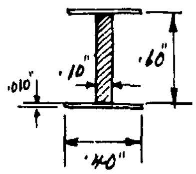
CENTRAL SPAR



$M = 534 \times 1.35 = 720$ lbs. ins. $S = 534$ lbs.

$q = \frac{534}{.6} = 890$ lbs/in.

SECTION



$$I_{NA} = 2 \times .40 \times .010 \times .3^2 + \frac{.60^3 \times .10}{12}$$

$$= .00072 + .0018 = \underline{.00252 \text{ in}^4}$$

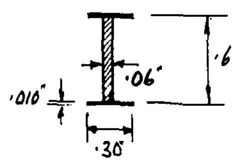
$$f = \frac{Mc}{I} = \frac{720 \times .3}{.00252} = 86,000 \text{ psi.}$$

$$Q = .004 \times .3 = .0012$$

$$\text{BRAZE SHEAR} = \frac{VQ}{I} = \frac{534 \times .0012}{.00252} = \underline{\underline{254 \text{ lbs/in.}}}$$

$$\text{BRAZE } f_s = 2,540 \text{ lbs/in}^2.$$

REDUCING THE SECTION TO .06



$$I = .006 \times .3^2 + \frac{.06 \times .6^3}{12}$$

$$= .00054 + .00108 = \underline{.00162 \text{ in}^4}$$

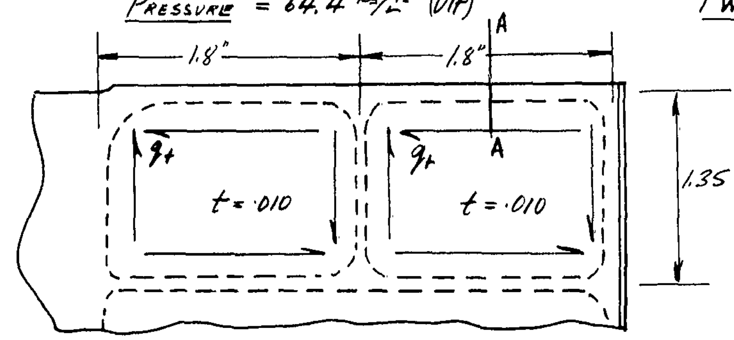
$$f = \frac{720 \times .3}{.00162} = 133,000 \text{ psi.}$$

$$\text{BRAZE SHEAR} = \frac{VQ}{I} = \frac{534 \times .003 \times .3}{.00162} = 297 \text{ lbs/in}$$

$$\text{BRAZE } f_s = \underline{\underline{4950 \text{ lbs/in}^2}}$$

$$f_s \text{ ON SPAR} = \frac{890 \text{ lbs/in}}{.06} = \underline{\underline{14800 \text{ lbs/in}^2}}$$

SKINS PRESSURE $= 64.4$ lbs/in² (ULT) FWD VANE



SHEARS q_t = TORQUE SHEAR $= 256$ lbs/in.

$b = 1.35$ $a = 1.8$ $b/a = .70$ $K = .70$

$E = 25 \times 1000^2$

$$f_{scr} = 25K\left(\frac{1000t}{b}\right)^2$$

$$= \underline{9600 \text{ lbs/in}^2} \qquad \tau/\tau_{cr} = 2.66$$

$$f_s = \underline{25{,}600 \text{ lbs/in}^2} \qquad \text{DIAG. TENS FACTOR } K = .20$$

$$\therefore w = \underline{.20 \times 256} = \underline{51.2 \text{ lbs/in.}}$$

PRESSURE STRESSES

(REF # 8 PAGE 224) $\frac{b}{t} = 1.35 \times 10^2$ $\left(\frac{b}{t}\right)^2 = 1.82 \times 10^4$

$\frac{p}{E} = \frac{64.4}{25} \times 10^{-6} = 2.58 \times 10^{-6}$ $\left(\frac{b}{t}\right)^4 = 3.32 \times 10^8$

$\frac{a}{b} = 1.33.$ $\frac{p}{E}\left(\frac{b}{t}\right)^4 = 855 \longrightarrow \frac{\sigma_3}{E}\left(\frac{b}{t}\right)^2 = 27$

(EXTRAPOLATED CONSERVATIVELY)

* MAX STRESS $\sigma_1 = \frac{60 \times 25 \times 10^6}{1.82 \times 10^4} = \underline{82{,}500 \text{ lbs/in}^2}$ @ 172,000 **M.S. HIGH.**

MEMBRANE STRESS $\sigma_3 = \frac{27 \times 25 \times 10^6}{1.82 \times 10^4} = \underline{37{,}000 \text{ lbs/in}^2}$

MEMBRANE PULL $= 370$ lb/in. $+ 51.2$ (T.F) $= 421$ lbs/in

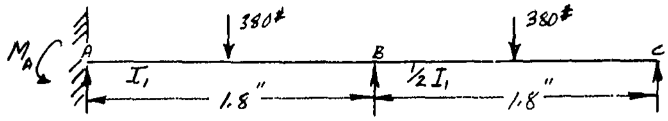
THIS IS MAX IN MID SPAN, DECREASING TO ZERO AT ENDS

TO SIMPLIFY, REPLACE BY EQUIV. POINT LOAD AT MID SPAN

$$W = \frac{1.8 \times 421}{2} = \underline{380 \text{ lbs}}$$

* σ_1 = MAX. STRESS — SIMPLY SUPPORTED EDGES

SKINS (CONT'D)



STIFFNESS FACTORS

AB $\frac{4EI_1}{L_1}$ $\frac{4EI_1}{L_1}$ 4 .727

BC $\frac{3EI_2}{L_2}$ $\frac{3EI_1}{2L_2}$ 1.5 .273

FIXING MOMENTS

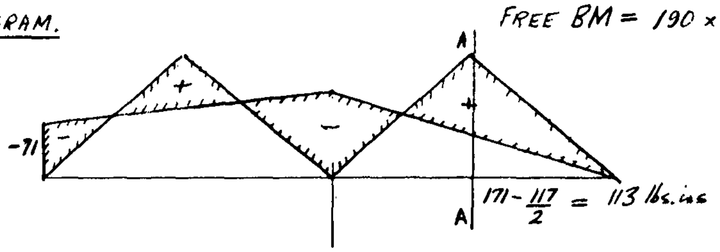
$$\frac{WL}{8} = \frac{380 \times 1.8}{8} = 86 \text{ lbs ins.}$$

HARDY-CROSS MOMENT DIST^N

A	.727 (B)	.273 (B)	C
-86	+86	-86	+86
		-43 ←	-86
-86	+86	-129	0
+15 ←	+31	+12	
-71	+117	-117	0

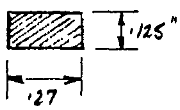
BM. DIAGRAM.

FREE BM = 190 × .9 = 171.



$171 - \frac{117}{2} = 113$ lbs.ins

SECTION AA.



$$Z = \frac{.125^2 \times .27}{6} = .00070 \text{ in}^3$$

$$\frac{M}{Z} = \frac{113}{.0007} = 160{,}000 \text{ psi.}$$

M.S. +.06

ASSUMING 1.10 PLASTIC BENDING FACTOR (CONSERVATIVE)
ALLOWABLE = 1.1 × 155,000 = 170,000

PIVOT MTG. FORWARD VANE

MOMENT = 830 lbs ins.

MIN SECTION ·375" O/D
·194" I/D.

$$\text{POLAR } I = \frac{\pi(D^4-d^4)}{32} = \frac{\pi}{32}(\cdot375^4-\cdot194^4)$$

$$I_p = \cdot001802 \qquad \frac{D}{2} = .1875$$

$$Z_p = \cdot0096 \text{ in}^3$$

$$f_s = \frac{T}{Z_p} = \frac{830}{\cdot0096} = \underline{86{,}500 \text{ lbs/in}^2}$$

$$\text{PIVOT LOAD} = \frac{1375}{2} = \underline{688 \text{ lbs}} \qquad (P.8)$$

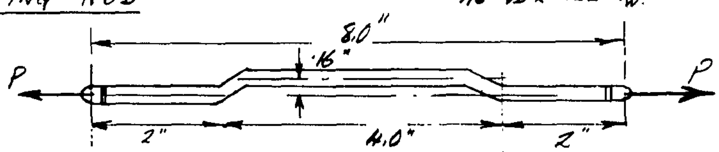
$$A = \cdot110 - \cdot029$$
$$= .081 \qquad f_s = \frac{688}{\cdot081}$$
$$= 8{,}500 \text{ psi}$$

$$\text{TOTAL } f_s = 86{,}500 + 8{,}500$$
$$= \underline{95{,}000} \text{ lbs/in}^2$$

M.S. +.05

$$\text{Allowable } f_s = 100{,}000 \text{ psi.}$$

CONNECTING ROD 3/16" OD x ·032" t_w.



$$P = 396 \text{ lbs.} \qquad A = \cdot0156 \text{ in}^2 \qquad \frac{P}{A} = 25{,}400 \text{ lb/in}^2$$

$e = \cdot16"$ $M = \cdot16 \times 396 = \underline{63.4 \text{ lbs. ins}}$ * SEE NEXT PAGE

$$I = \frac{\pi}{64}\{D^4-d^4\} = \frac{\pi}{64}\{\cdot187^4 - \cdot125^4\} = \underline{\underline{\cdot00005 \text{ in}^4}}$$
$$(50 \times 10^{-6};$$

CONNECTING ROD

SECTION MODULUS $= \dfrac{\cdot 00005}{\cdot 0935} = \cdot 000534$

$$f = \frac{P}{A} + \frac{M}{Z} = 25{,}400 + 105{,}000$$

$* M = 396(\cdot 16 - \cdot 02) = 56$ lbs ins
SEE BELOW

$f = \underline{130{,}400 \text{ psi.}}$
@ 154,000

M.S. +.18

DEFLECTION UNDER LIMIT LOADS

CENTER PORTION $M = 40$ lbs ins (I Constant)
$\ell = 4''$

1st Approx

$$\sum_0^{\ell/2} M dx = 40 \times 2 = 80$$

$$\Sigma\Sigma M\,dx\,dx = \frac{80}{2} \times 2 = 80$$

$$\delta = \frac{80}{EI} = \frac{80}{25 \times 50} = \cdot 064$$

M would decrease to $\cdot 10 \times 200 = 20$ lbs. ins

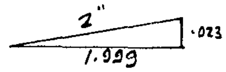
2nd Approx $\Sigma M = 20$ $\Sigma\Sigma M = 20$ $\delta = \cdot 016''$

3rd Approx $M = 30$ $\Sigma\Sigma M = \dfrac{30}{25 \times 50} = \cdot 024$ $M = 25$ lbs ins,

4th Approx $M = 28$ $\delta = \dfrac{28}{25 \times 50} = \cdot 022$ $M = 27.6$ lbs ins (LIMIT)

$\delta = \cdot 023''$ Approx. $\theta = \dfrac{\cdot 023}{2} = \cdot 0115$ rad. $\cdot 66°$

* (REDUCES M IN CENTRE)

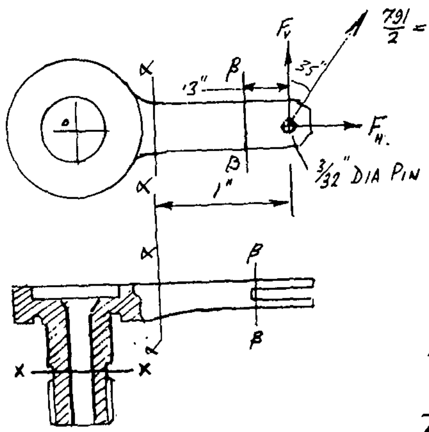


$$\text{Total stretch} = \frac{P\ell}{AE} + 2 \times \cdot 001$$

$$= \frac{200 \times 8}{15600 \times 25} + \cdot 002$$

$$= \cdot 004 + \cdot 002 = \underline{\underline{\cdot 006''}}$$

LEVER — AFT VANE



$A \quad \frac{791}{2} = 396$ lbs (REF. Page 3)

$F_V = 324$ lbs
$F_H = 227$ lbs

$M_{\alpha\alpha} = 324$ lbs.ins

$Z_{\alpha\alpha} = \frac{.2 \times .3^2}{6} = .003$

$\frac{M}{Z} = \underline{108,000 \text{ lbs/i}^2}$

$\frac{P}{A} = \frac{227}{.06} = \underline{3,800 \text{ lbs/i}^2}$

$M_o = 324 \times 1.5 = 485$ lbs.ins

TOTAL $= \underline{111,800 \text{ lbs/i}^2}$ ÷ 150,000 M.S. +.34

SECTION ββ. $M = .3 \times 324 =$ 97.2 lbs ins
48.6 lb ins per lug.

$Z = \frac{.31^2 \times .04}{6} = \underline{.00064 \text{ i}^2}$ per lug.

$\frac{M}{Z} = \frac{48.6}{.00064} = \underline{76,000 \text{ lbs/i}^2}$

$\frac{P}{A} = \frac{227}{2 \times .04 \times .31} = \underline{9,100 \text{ lbs/i}^2}$

TOTAL $= \underline{85,100 \text{ lbs/i}^2}$ ÷ 150,000 M.S. HIGH

SECTION XX — TORQUE = 485 lbs.ins

$Z_p = .0096$ (REF P.19)

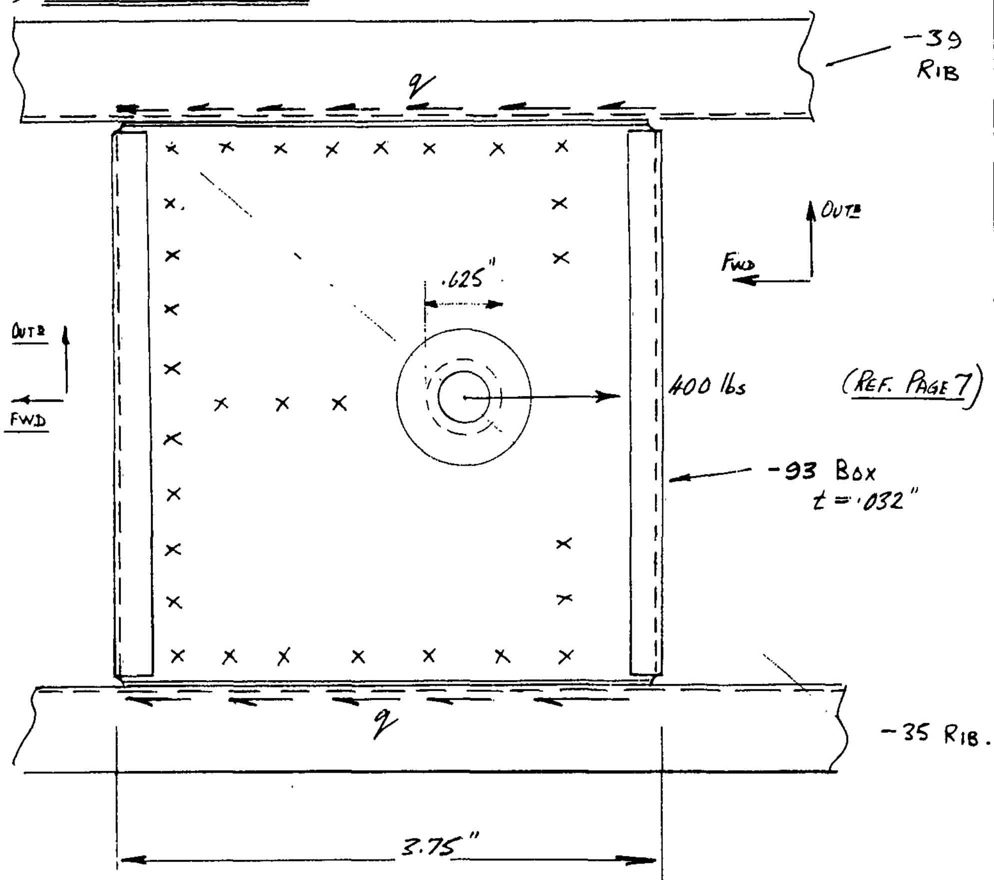
$f_s = 50,500$ lbs/i²

M.S. HIGH — DIAS. MADE THE SAME AS THE FWD MTG. TO KEEP SAME SIZE OF SPLINES (SIMPLICITY OF MACHINING)

$\frac{3}{32}$" DIA PIN
REF # 6 PAGE 212 SINGLE SHEAR 518 DS. = 910 = 637 lbs
TEMP RED .7 M.S HIGH

TIP STRUCTURE

a) FWD VANE PIVOT



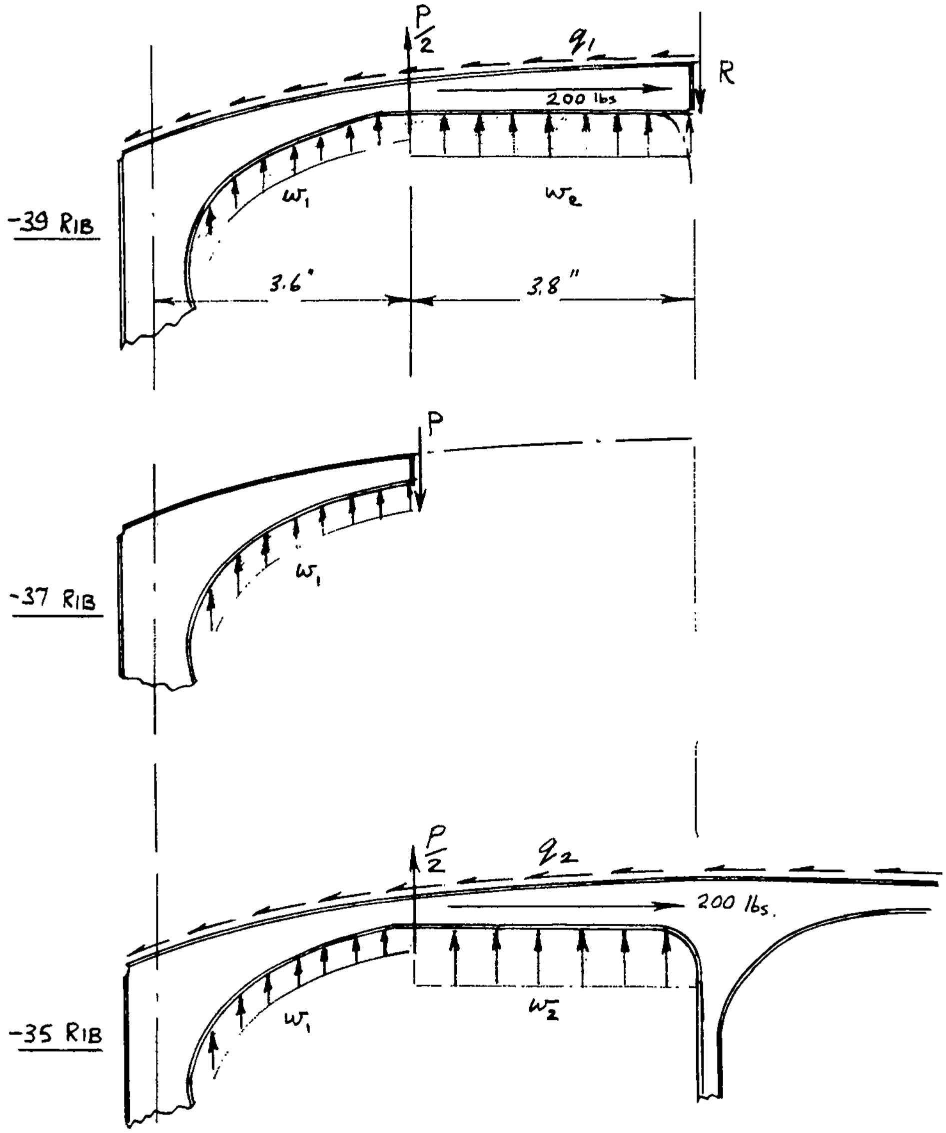
$$q = \frac{200}{3.75} = 53 \ \text{lbs/in.}$$

BEARING OF BUSH

$$f_{brg} = \frac{400}{.125 \times .032} = 20{,}000 \ \text{psi.}$$ M.S. HIGH

$$f_s = \frac{53}{.032} = \underline{1650 \ \text{psi.}}$$

LOADS TO RIBS TIP – FWD PORTION



LOADS ON OUTER RIBS (CONT'D)

-39 RIB WILL BE MOST CRITICAL OF THREE (PROPPED CANTLV)

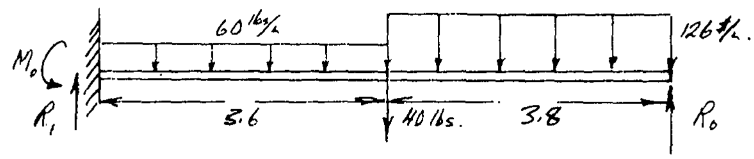
LIMIT PRESSURE (REF # 7) = 32 psi   LOAD FACTOR 1.5

$1.5 \times 32 = 48$ psi.

$w_1 = 48 \times 1.25 = 60$ lbs/in
$w_2 = 48 \times (.625 + 2.00) = 126$ lbs/in.

$q_1 = \frac{200}{7.4} = 27$ lbs/in.

$P = \frac{66 \times 3.6 \times 3}{8} = 81$ lbs $\frac{P}{2} = 40$ lbs (Appr.)



APPROXIMATION $60 \times 7.4 = \underline{444 \text{ lbs.}}$ $66 \times 3.8 = \underline{251 \text{ lbs}}$

$$M_o = \frac{444 \times 7.4}{8} + \frac{3}{16} \times 40 \times 7.4 + \frac{251}{2} \frac{5.5 \times 1.9}{7.4}\left(2 - \frac{5.5}{7.4}\right)$$

$$= 411 + 55.6 + 223 = \underline{690 \text{ lbs.ins}}$$

$$\therefore R_o = \frac{444 \times 3.7 + 251 \times 5.5 + 40 \times 3.6 - 690}{7.4}$$

$$= \frac{1640 + 1380 + 144 - 690}{7.4} = \frac{2474}{7.4} \frac{\text{lbs ins}}{\text{in}}$$

$$R_o = \underline{334 \text{ lbs}}$$

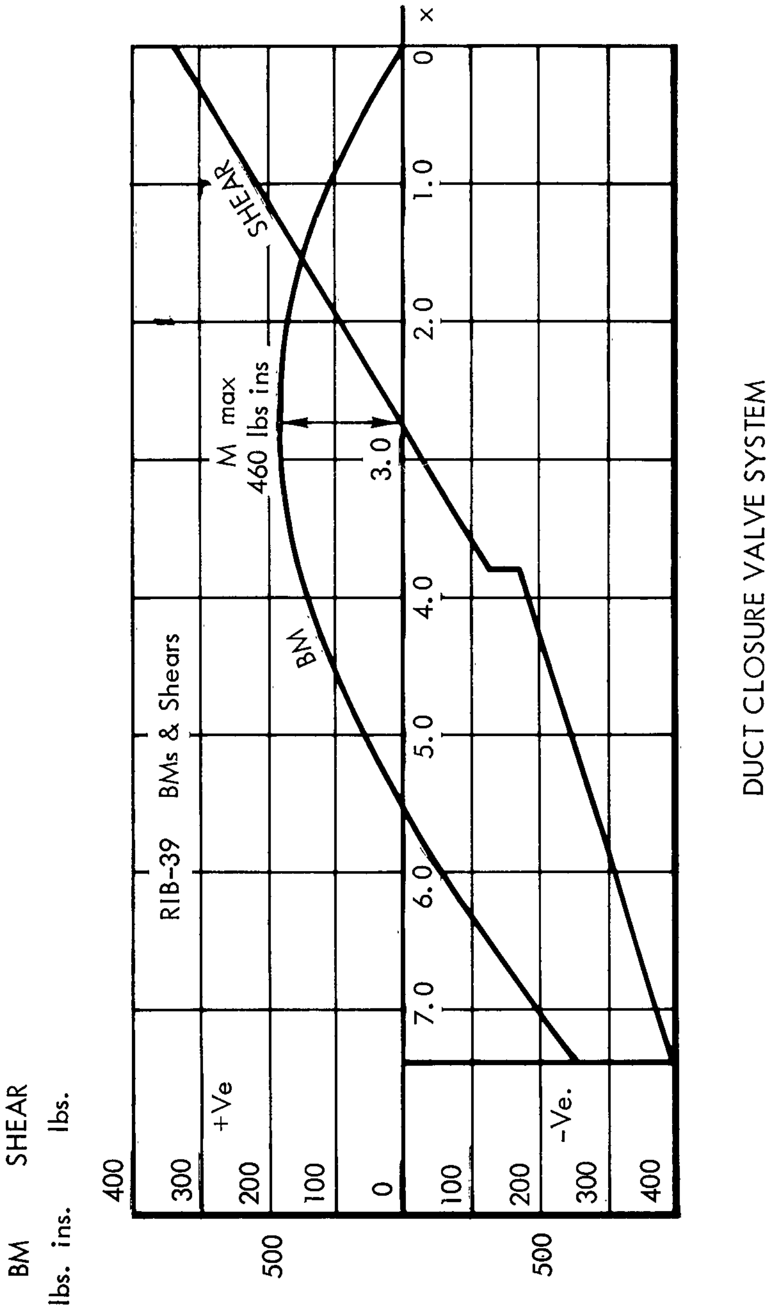
CHECK $R_o = \frac{3}{8} \times 444 + \frac{5}{16} \times 40 + \frac{251}{2} \times \left(\frac{5.5}{7.4}\right)^2 \left(3 - \frac{5.5}{7.4}\right)$

$$= 167 + 13 + 169 = 349 \text{ lbs.}$$

USE $R = 340$ lbs.
& $M = 690$ lbs.ins

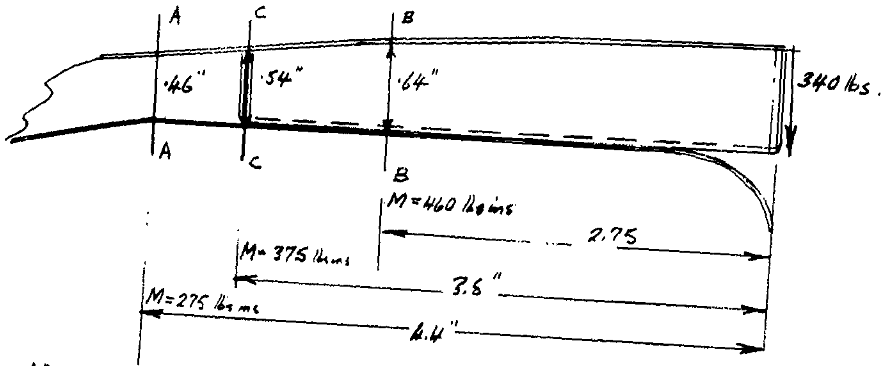
MAX EL DUE TO q_1 & 200 lbs

$F = 200 - 27 \times 3.8 = \underline{100 \text{ lbs (Approx)}}$
Tension



Duct Closure Valve System

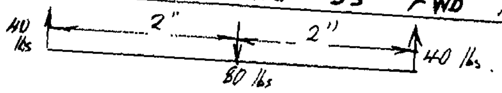
-39 RIB CONT'D



$$\text{MAX } q = \frac{340}{.7} = 486 \text{ lbs/in} \qquad t = .020$$

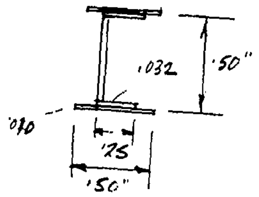
$$f_s = 24{,}300 \text{ lbs/in}^2$$

SPANWISE BM ACROSS -93 FWD FLANGE



$$M = 40 \times 2 = 80 \text{ lbs ins}$$

SECTION (MID SPAN)



$$\frac{M}{d} = 160 \text{ lbs}$$

$$A = .25 \times .032 + .50 \times .010$$

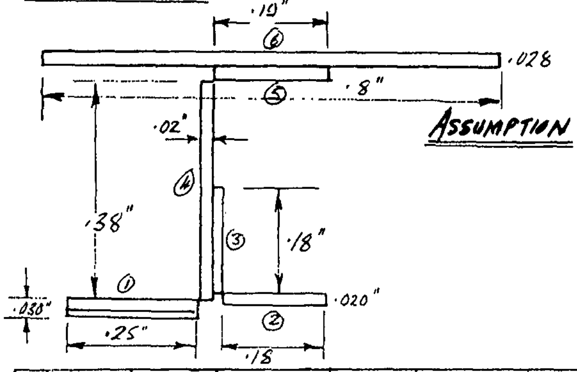
$$= .008 + .005 = .013$$

$$f = \frac{160}{.013} = 12{,}300 \text{ lbs/in}^2$$

M.S. HIGH.

−39 RIB CONT'D.

SECTION AA. $M = 275$ lbs ins.



ASSUMPTION

THERMAL STRESSES IN ① & ② SAME AS FOR SECTION BB.
$= 50{,}000$ lbs/in².

#	a	b	A	h	Ah	y	y²	Ay²	ab³/12
1	·250	·030	·0075	·015	·000112	·269	·072	·00054	—
2	·200	·020	·0040	·035	·00014	·249	·062	·00025	—
3	·020	·180	·0036	·120	·00043	·164	·027	·00010	—
4	·020	·380	·0076	·220	·00167	·064	·004	·00003	·00009
5	·190	·020	·0038	·420	·0016	·136	·0184	·00007	—
6	·800	·028	·0224	·444	·010	·160	·0255	·00057	—
			·0469	·284	·013952			·00156	·00009

$y_c = ·284$ $\quad Z_c = ·0058 \text{ in}^3$

TOTAL $I = ·00165 \text{ in}^4$

$$\frac{M}{Z} = \frac{275}{·0058} = 47{,}400 \text{ lbs/in}^2$$

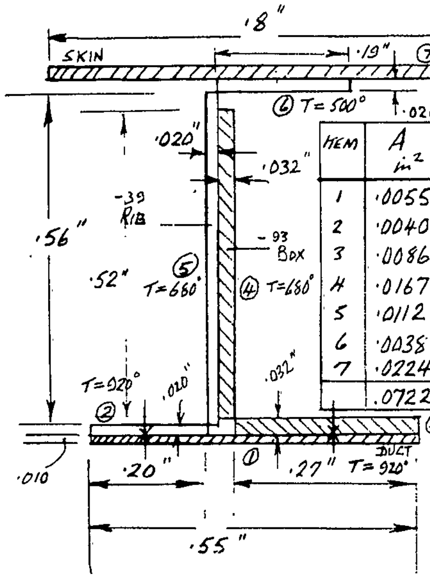
TOTAL $f_c = 47{,}400 + 50{,}000$

$= 97{,}400$ lbs/in². M.S. +.09

ALLOWABLE 106,000 lbs/in²
REF #11 p. 5.2.5.10.

-39 RIB (CONT'D)

SECTION B-B (MAX BM) $M = 460$ lbs ins



THERMAL STRESSES.

ITEM	A in^2	ΔT	$\alpha \times 10^{-6}$	$E \times 10^6$	f'	ΔP	$f'' = \frac{\Sigma \Delta P}{\Sigma A}$	f_{RES} #/in²
1	.0055	850	7.8	27	179,000	985		-49,000
2	.0040	850	"	27	179,000	716		-49,000
3	.0086	850	"	27	179,000	1540	130,000	-49,000
4	.0167	610	"	29	135,000	2300		-8,000
5	.0112	610	"	29	135,000	1550		-8,000
6	.0038	430	"	30	100,000	380		+30,000
7	.0224	370	9.4	25	87,000	1950		+43,000
	.0722					9421		

THESE STRESSES ADD TO PRESSURE BENDING

PRESSURE STRESSES

#	a	b	A	h	Ah	y	y^2	Ay^2	$ab^3/12$
1	.55	.010	.0055	.005	.00003	.343	.118	.00065	—
2	.20	.020	.0040	.020	.00008	.328	.108	.00043	—
3	.27	.032	.0086	.026	.00022	.322	.104	.00089	—
4	.032	.520	.0167	.302	.00505	.046	.0021	.00003	.000376
5	.020	.560	.0112	.310	.00347	.038	.0014	.00002	.000292
6	.190	.020	.0038	.600	.00228	.252	.0634	.00024	—
7	.800	.028	.0224	.624	.01400	.276	.0760	.00170	—
			.0722	.348	.02513			.00396	.000668

$y_t = .292$ $\quad Z_t = .0158$ $\qquad$ TOTAL $I = .004628$

$y_c = .348$ $\quad Z_c = .0133$

$$f_{t_{max}} = \frac{460}{.0158} + 43{,}000 = 72{,}100 \text{ lbs/in}^2$$

$$f_{c_{max}} = \frac{460}{.0133} + 49{,}000 = 83{,}600 \text{ lbs/in}^2$$

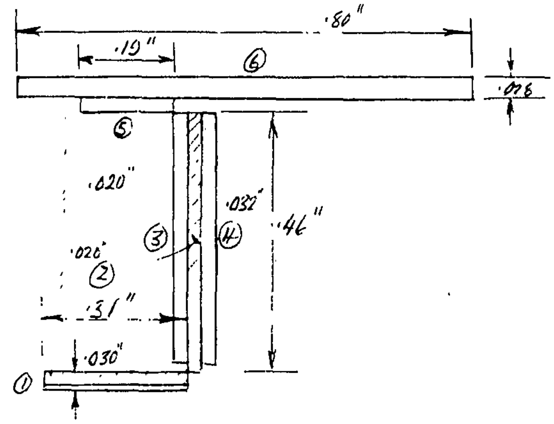
$f_{c_{ALL.}} = 106{,}000$ lbs/in² $\qquad$ M.S. +.27

LOCAL INSTABILITY

REF # 11 p. 5.2.5. IC

-39 RIB (CONT'D)

SECTION C-C — (REVISED) M = 375 lbs ins



ASSUME THERMAL STRESSES = 50000 psi (in ELEMENT 1)

#	a	b	A	h	Ah	y	y^2	Ay^2	$ab^3/12$
1	.31	.030	.0091	.015	.000136	.312	.097	.00088	—
2	.02	.46	.0092	.260	.002390	.067	.0045	.00004	.00016
3	.02	.43	.0086	.260	.002240	.067	.0045	.00004	.00013
4	.032	.43	.0138	.260	.00359	.067	.0045	.00006	.00021
5	.19	.02	.0038	.500	.0019	.173	.030	.00011	—
6	.80	.028	.1224	.520	.0116	.193	.0371	.00083	—
			.0669	.327	.021856			.00196	.00050

$y_c = .327$

$Z_c = .0075$

TOTAL $I = .00246 \text{ in}^4$

$\frac{M}{Z} = 50{,}000$ lbs/in²

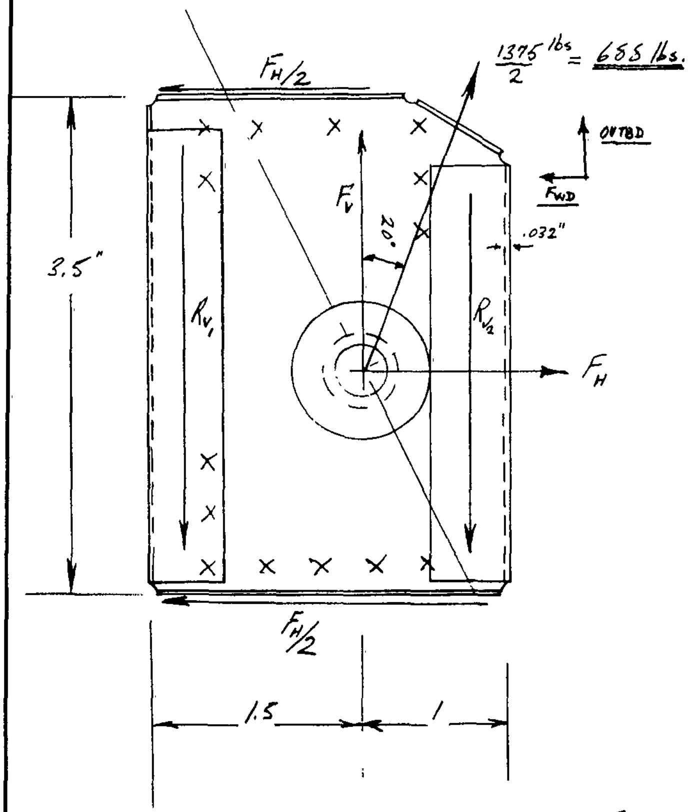
TOTAL f_c = 100,000 lbs/in² MS. +.06

ALLOWABLE 106,000 lbs/in²

(LOCAL INSTABILITY)

REF # 11 p. 5.2.5.10

PIVOT MOUNTING — AFT. VANE



$$F_V = 688 \cos 20°$$
$$= 688 \times .9397$$
$$= 645 \text{ lbs.}$$

$$F_H = 688 \sin 20°$$
$$= 688 \times .342$$
$$= 235 \text{ lbs.}$$

$$F_H/2 = 118 \text{ lbs.}$$

$$R_{V_1} = \frac{1.5}{2.5} \times 645$$
$$= \underline{\underline{387 \text{ lbs}}}$$

$$R_{V_2} = \underline{\underline{258 \text{ lbs}}}.$$

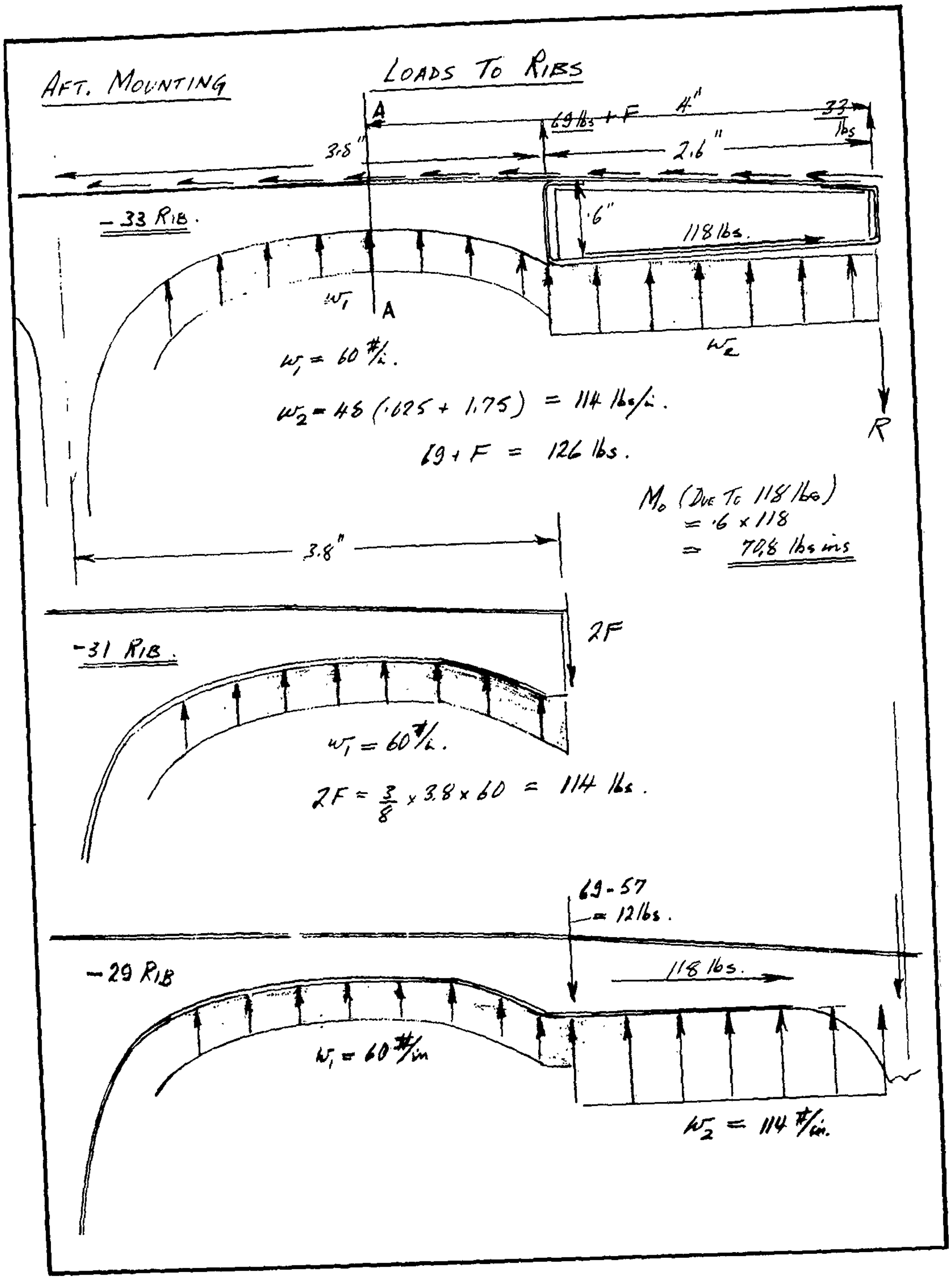
KICK LOADS DUE TO TRANSFER OF F_V TO SKIN. d = .62 for R_{V_1}, .45 for R_{V_2}

$$M_1 = R_{V_1} \times .62 = 240 \text{ lbs ins}$$

$$P_1 = \frac{240}{3.5} = \underline{\underline{69 \text{ lbs}}}$$

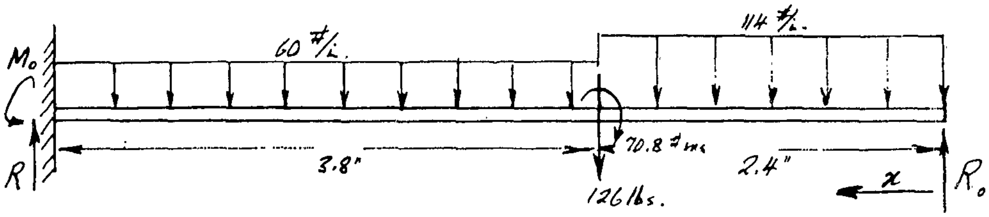
$$M_2 = R_{V_2} \times .45 = 116 \text{ lbs. ins}$$

$$P_2 = \frac{116}{3.5} = \underline{\underline{33 \text{ lbs}}}$$


AFT. MOUNTING
LOADS TO RIBS
A
69 lbs + F
4"
33 lbs
3.8"
2.6"
-33 RIB.
.6"
118 lbs.
w1
A
w2
w1 = 60 #/in.
w2 = 48 (.625 + 1.75) = 114 lbs/in.
69 + F = 126 lbs.
R
Mo (Due To 118 lbs)
= .6 x 118
= 70.8 lbs ins
3.8"
2F
-31 RIB.
w1 = 60#/in.
2F = 3/8 x 3.8 x 60 = 114 lbs.
69 - 57
= 12 lbs.
118 lbs.
-29 RIB
w1 = 60 #/in
w2 = 114 #/in.

AFT MTG — RIB LOADING CONT'D.

-33 RIB MOST CRITICAL OF THE THREE



APPROXIMATION

$114 - 60 = 54 \text{ \#/in} \times 2.4 = 130 \text{ lbs.}$

$60 \times (3.8 + 2.4) = 372 \text{ lbs.}$

$$M_0 = \frac{372 \times 6.2}{8} \qquad 288 \text{ lbs ins}$$

$$+ \frac{126}{2} \times \frac{3.8 \times 2.4}{6.2}\left(2 - \frac{3.8}{6.2}\right) \qquad 128 \text{ "}$$

$$+ \frac{130}{2} \times \frac{5 \times 1.2}{6.2}\left(2 - \frac{5.6}{6.2}\right) \qquad 75 \text{ "}$$

$$- \frac{70.8}{2}\left[1 - 3\left(\frac{2.4}{6.2}\right)^2\right] \qquad -19 \text{ "}$$

TOTAL $M_0 = 472$ lbs. ins

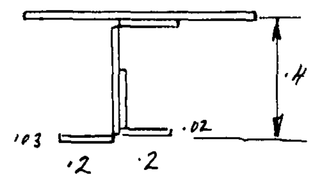
$$6.2 R_0 = (372 \times 3.1) + (126 \times 3.8) + 70.8 + (130 \times 5.0) - 472$$

$$= 1152 + 480 + 71 + 650 - 472 = 1881$$

$R_0 = \underline{304 \text{ lbs}}$ $\quad R_1 = \underline{324 \text{ lbs}}$

BM. DIAGRAM PLOTTED ON NEXT PAGE

SECTION AA $\quad M = 110$ lbs ins. $\qquad$ THERMAL STRESSES $f_c = 60,000$ psi (CONSERVATIVE)



$$\frac{M}{d} = \frac{110}{.4} = 276 \text{ lbs}$$

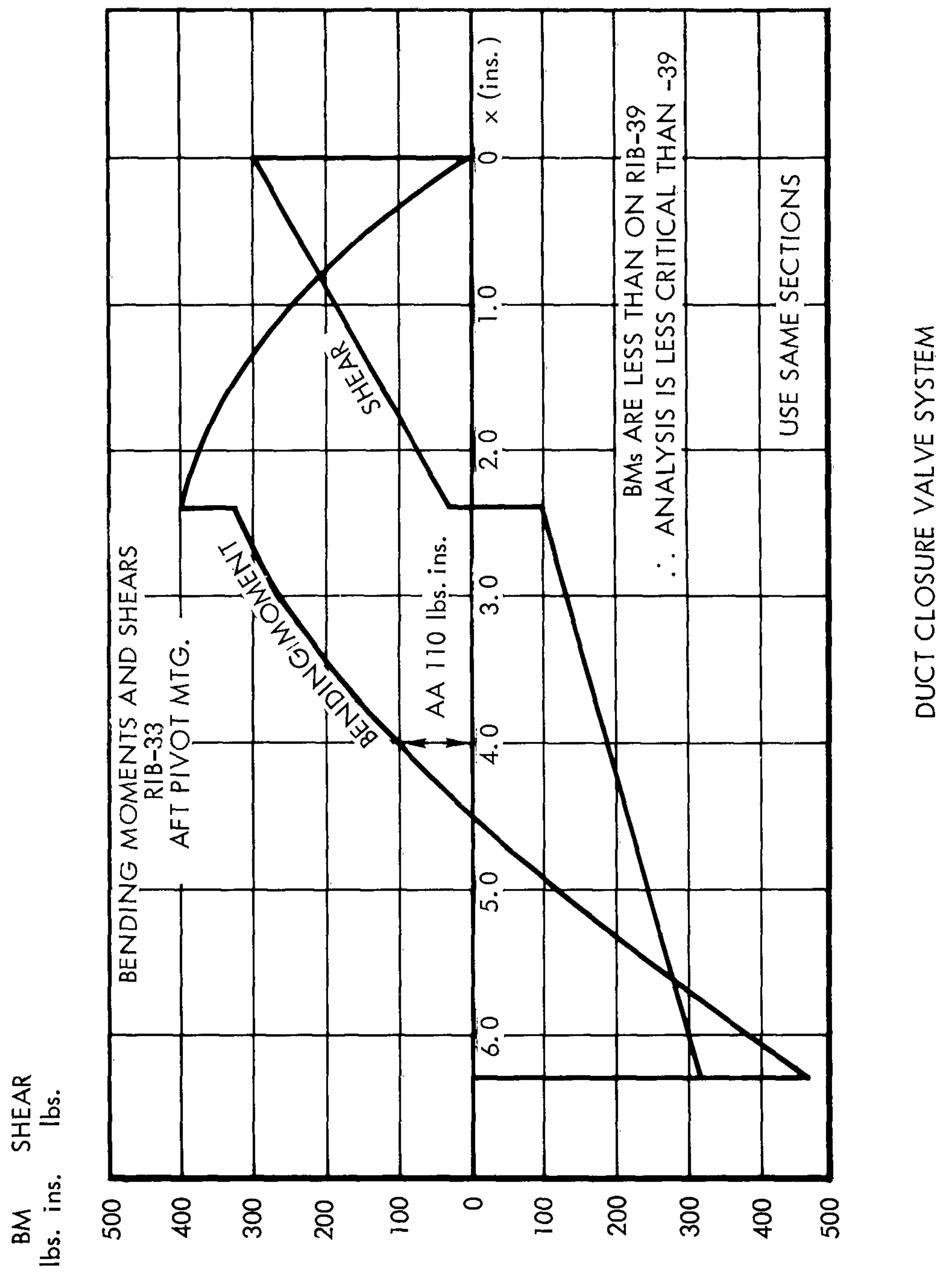
$$A = .02 \times .2 + .03 \times .2$$

$$= .004 + .006 = .010$$

$$f = 27,600 \text{ lbs/in}^2$$

TOTAL $= 87,600$ lbs/in²

NOT CRITICAL



Duct Closure Valve System

PRESSURE PANEL FORMED BY BOX-93 & DUCT

PANEL $3.5 \times 4.0''$ $\quad t = .010 + .032 = .042''$

REF. #8 PAGE 224

$\frac{b}{t} = \frac{3.5}{.042} = 83.4$ $\quad p = 64.4\ \text{lbs/in}^2$

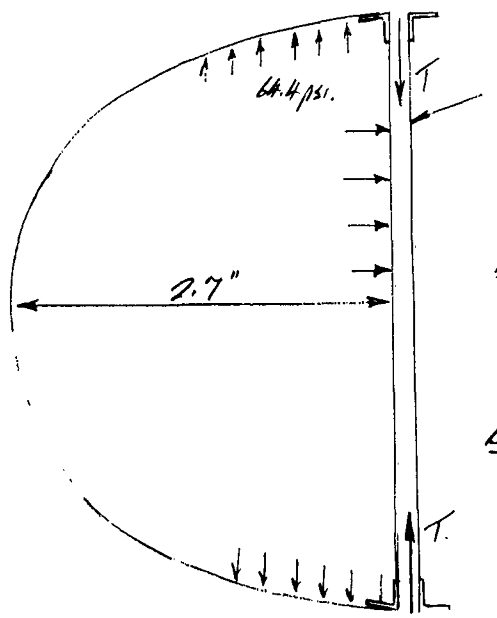
$\left(\frac{b}{t}\right)^2 = 69.2 \times 10^2$ $\quad \left(\frac{b}{t}\right)^4 = 48 \times 10^6$

$\frac{p}{E}\left(\frac{b}{t}\right)^4 = \frac{64.4 \times 48 \times 10^6}{25 \times 10^6} = 124$

$\frac{a}{b} = 1.0$ $\quad \frac{\sigma_2}{E}\left(\frac{b}{t}\right)^2 = 32$

$\sigma_2 = \frac{32 \times 25 \times 10^6}{6.92 \times 10^3} = \frac{125{,}000\ \text{psi.}}{@\ 154{,}000}$ $\quad$ M.S. $\underline{+.33}$

-61 FIXED VANE



$T = \frac{64.4 \times 2.7}{2} = 87\ \text{lbs/in.}$

$f_t = \frac{87}{.02} = 4300\ \#/\text{in}^2$

PANEL SIZE $4'' \times 1''$

MEMRANE TENSION = 56,000 lbs/in² (APPROX)

(REF. PAGE) OK

BENDING OF STIFFENERS :-

VDL = 36 lbs/in (L.F. 1.5)

$\ell = 4''$

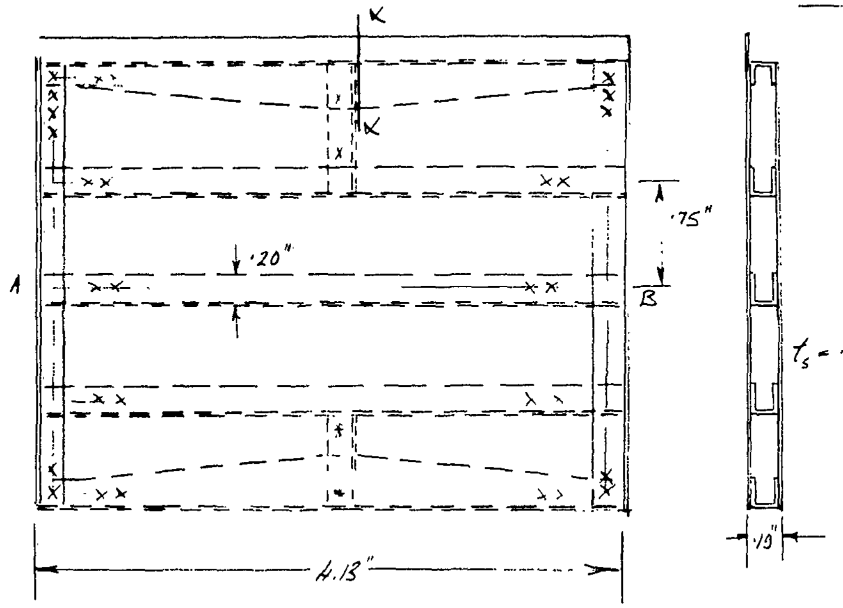
$\frac{w\ell^2}{8} = \frac{36 \times 4 \times 4}{8}$

$= \underline{72\ \text{lbs ins}}$

FIXED VANE – 61

PRESSURE = 32 psi LIMIT
= 48 psi ULT.

LOAD FACTOR 1.5
(COMBINING WITH THERMAL STRESS)

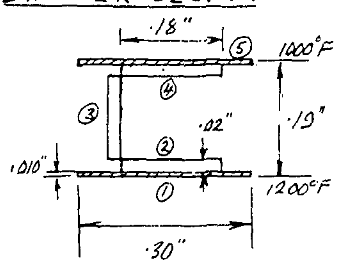


LOAD ON STIFFENER A-B.

$$.75 \times 48 = 36 \text{ lbs/in}$$

$$M = \frac{wl^2}{8} = \frac{36 \times 4.13^2}{8} = \underline{76.9 \text{ lbs. ins}}$$

STIFFENER SECTION



ITEM #	A	y	y^2	Ay^2	$ab^3/12$
1	.003	.090	.0081	.0000243	—
2	.0036	.075	.0056	.0000202	—
3	.0026	0	—	—	.0000037
4	.0036	.075	.0056	.0000202	—
5	.003	.090	.0081	.0000243	—
	.0158			.0000890	.0000037

$y = .095$

TOTAL $I = .0000927 \text{ in}^4$

$Z = .000978 \text{ in}^3$

$$\frac{M}{Z} = \underline{79{,}600 \text{ lbs/in}^2}$$

-61 – FIXED VANE (CONT'D)

THERMAL STRESSES ON STIFFENER

#	A	ΔT	α	E	f'	ΔP	$f = \frac{\Sigma \Delta P}{\Sigma A}$	f_{RES}
1	.003	1130	↑	↑	220,000	660	↑	−20,000
2	.0036	1130			220,000	794		−20,000
3	.0026	1030	7.8	25	201,000	523	200,000	0
4	.0036	930			181,000	651		+19,000
5	.003	930	↓	↓	181,000	544	↓	+19,000
	.0158					3172		

$MAX\ f_c = 79,000 + 20,000 = 99,000\ lbs/in^2$

$\frac{b}{t} = \frac{.18}{.02} = 9$

$f_{cr} = 120,000$

Temp Red Factor = .86

$f_{allowable} = 103,000\ lbs/in^2$ $\qquad$ M.S +.04

(REF. #8 PAGE 224)

PRESSURE LOADS IN SKINS $\quad b = .75''$ $\quad a = 2.0''$ $\quad \frac{a}{b} = 2.67$

$\frac{b}{t} = 75 \qquad \left(\frac{b}{t}\right)^2 = 5.6 \times 10^3 \qquad \left(\frac{b}{t}\right)^4 = 31.3 \times 10^6$

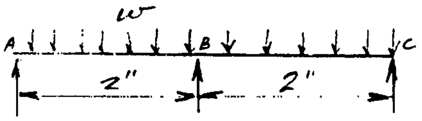
LF = 2.0 $\qquad \frac{p}{E}\left(\frac{b}{t}\right)^4 = \frac{64}{25} \times 31.3 = 80$

REF. PAGE. $\qquad$ Max Stress $\frac{\sigma_2}{E}\left(\frac{b}{t}\right)^2 = 32$

$$\sigma_2 = \frac{32 \times 25 \times 10^6}{5.6 \times 10^3} = 143,000\ lbs/in^2$$

MEMBRANE STRESS $\quad \frac{\sigma_3}{E}\left(\frac{b}{t}\right)^2 = 3$

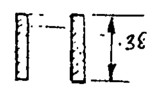
$$\sigma_3 = \frac{25 \times 3 \times 10^6}{5.6 \times 10^3} = 13,400\ lbs/in^2$$



$w = 134\ lb/in \qquad W = 2 \times 134 = 268$

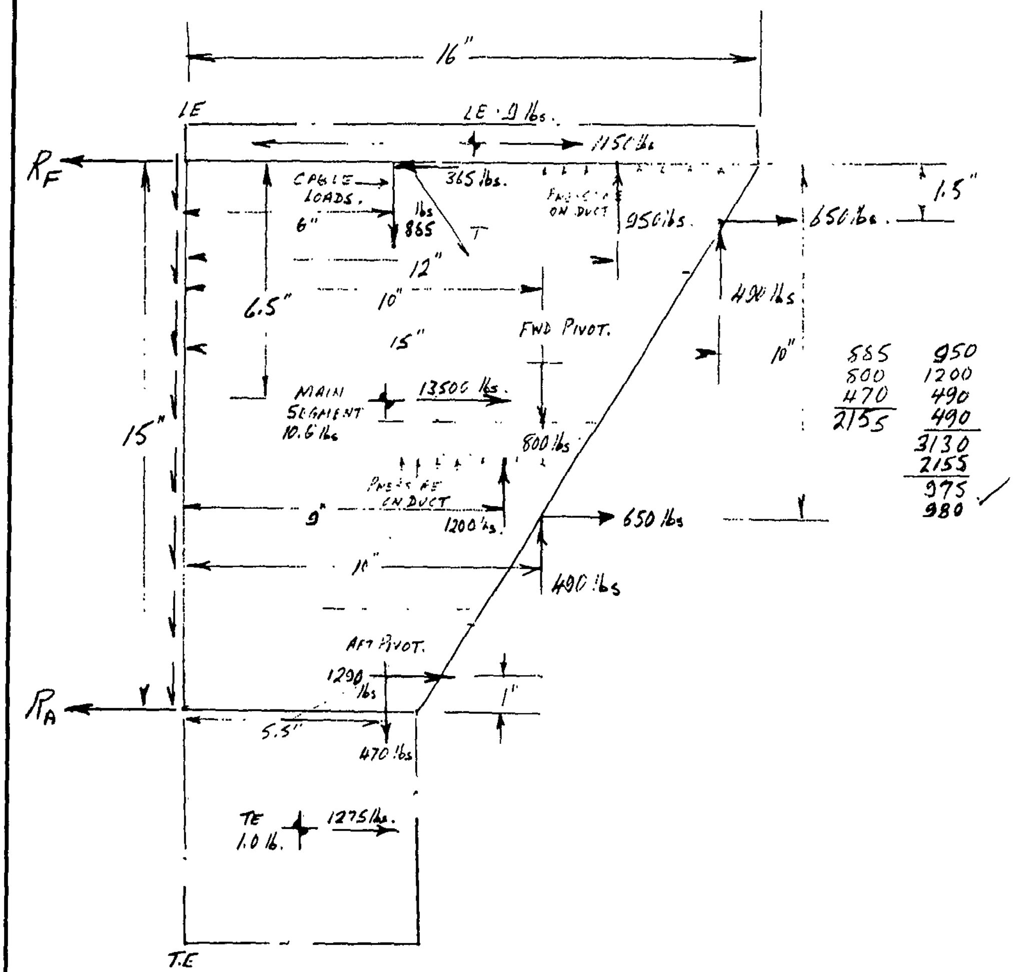
$M_B = \frac{W\ell}{8} = \frac{268 \times 2}{8} = 67\ lbs\ in.$

SECTION KK



$Z = \frac{.38^2 \times .03}{6} = .000703 \qquad \frac{M}{Z} = 95,000\ lbs/in^2$

LOADS ON TIP.   REFERENCE No 11 PAGE 5.2.5.6



ITEM.	WT.	ULT. INERTIA g's.	ULT FORCE
CASCADE	5.75	1275	7330 lbs
PRESSURE	—	—	1300 lbs.
MAIN SEGMENT	10.6	1275	13,500 lbs
TRAILING EDGE	1.0	1275	1,275 lbs
LEADING EDGE	.9	1275	1,150 lbs

TOTAL FWD $= \frac{1960}{2} = 980$ lbs   TOTAL $^A/_B = \frac{2600}{2} = 1300$.

LOADS ON TIP (CONT'D)

MOMENTS ABOUT F/SPAR

+ve M. F_y FWD, +Ve LOADS. F_x O/B

a) CENTRIFUGAL

	F_y	x	M	F_x	y	M
CASCADE	—	—	—	7,330 lbs	7.5	55,000
MAIN SEGMENT	—	—	—	13,500 lbs	6.5	87,900
TRAILING EDGE	—	—	—	1275 lbs	18	23,000
LEADING EDGE.	—	—	—	1150 lbs	7.0	−1,150
				34,255		164,750

b) PRESSURE (VANE LOADS ETC)

ITEM	F_y	x	M	F_x	y	M
AFT. PIVOT	−470	5.5	+2,590	+1290	14	+18,100
FWD. PIVOT	−800	10.0	−8,000	—	—	—
AFT. DUCT	+1200	9.0	+10,800	—	—	—
FWD. DUCT	+950	12.0	+11,400	—	—	—
AFT. CASCADE	+490	10.0	+4,900	+650	10	+6,500
FWD. CASCADE	+490	15.0	+7,350	+650	1.5	+975
CABLE LOADS.	−885	6.0	−5,310	−365	0	—
	+975		+23,730	+2225		+25,575

$$\text{TOTAL } M = 23{,}730 + 25{,}575 + 164{,}750$$

$$= \underline{214055 \text{ lbs ins}}$$

$$R_A = \frac{214055}{15} = \underline{14{,}300 \text{ lbs.}}$$

$$R_F = 34{,}255 + 2225 - 14{,}300 \text{ lbs.}$$

$$= \underline{22{,}180 \text{ lbs}}$$

$$\text{MAX SKIN } q = \frac{14{,}300}{2 \times 6.5} = 1100 \text{ lbs/in} \quad (\text{per. skin})$$

$t = .028$

$$f_s = \underline{39{,}200 \text{ lbs/in}^2}$$

$$f_{\text{allowable}} = 58{,}400 \text{ psi.}$$

M.S. +.50

LOADS ON TIP (CONT'D)

AFT WEB

$q_{max} = 1100$ lbs/in.

$t = .020$

$f_s = 55,000$ psi.

$f_{su} = 58,400$

M.S. +.06

ATTACH TO SPAR

$P = 14,300$ lbs

3 - 5/16 BOLTS

LOAD/BOLT = 4,767 lbs.

FROM REF # 11 PAGE 5.2.59

$$f_{bry} = \frac{4767}{.313 \times .071} = 214,000 \text{ lbs/in}^2$$

REF # 11 PAGE 5.2.5.9

$T = 460°$

M.S. 0.00

$$f_{bru} = 300 \times .71 = 214,000 \text{ lbs/in}^2$$

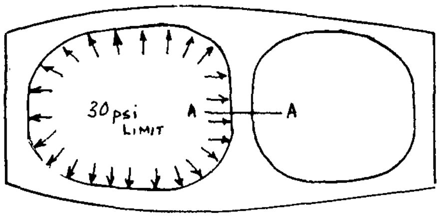
(REF # 6 PAGE.35)

DUCT PRESSURE AND THERMAL ANALYSIS (INBD VALVE CONCEPT)

THE FOLLOWING ANALYSIS IS BASED ON THE CONCEPT OF A CLOSURE VALVE LOCATED AT THE INBD END OF THE DUAL DUCTS. THIS CONCEPT WAS ABANDONED IN FAVOR OF THE TIP VALVES, BECAUSE OF EXCESSIVE STRESS IN THE STRUCTURE AS INDICATED BELOW.

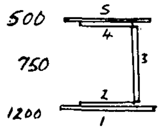
RIB ANALYSIS

i) CONSTANT SECTION.



TEMPERATURE DIFFERENTIAL ACROSS CENTER POST SECTION AA
700 °F
(1200 → 500 °F)

SECTION A·A.



THERMAL STRESSES

#	a	b	ΔA	ΔT	f′	ΔP	f″	f_R psi.
1	·3	·010	·0030	1130	−226,000	680		−74,000
2	·18	·010	·0018	1130	−226,000	407		−74,000
3	·010	·280	·0028	680	−136,000	380	+152,000	16,000
4	·18	·010	·0018	430	−86,000	155		+66,000
5	·30	·010	·0030	430	−86,000	258		+66,000
			.0124			1880		

$f' = \Delta A \times \Delta T \times \alpha \times E$

$\alpha = 7.4 \times 10^{6}$

$E = 27 \times 10^{6} \quad \alpha E = 200$

PRESSURE BENDING $\quad M_{AA} = \frac{w l^2}{24} = \frac{30 \times 1.33 \times 3^2}{24} = 15$ lbs.ins

$P_{couple} = \frac{15}{.30} = 50$ lbs.ins $\quad A = (.3 + .19) \times .01 = .0049\ in^2$

$f = \underline{\underline{10{,}200\ psi.}}$

TOTAL = 84,200 #/in² (LIMIT) $\quad$ FLANGE INST. $b/t = 19$

$f_{cr} = 66{,}000$ AT RT

$= 57{,}000$ psi AT 1200 °F

FLANGE WOULD CRIPPLE UNDER LIMIT LOADS

ii) TRANSITION AREA

FROM ANALYSIS SIMILAR TO CONSTANT SECTION (ABOVE)

$f_{THERMAL} = 65{,}000$ psi.

$f_{PRESSURE} = \underline{43{,}200}$ psi

$108{,}200$

FLANGE CRIPPLING = 66,000 psi 1200 °F

FLANGE WOULD BUCKLE UNDER LIMIT LOADS

DISTRIBUTION

USCONARC	3
First US Army	3
Second US Army	2
Third US Army	2
Fourth US Army	1
Sixth US Army	1
USAIC	2
USACGSC	1
USAWC	1
USAATBD	1
USAARMBD	1
USAAVNBD	1
USATMC(FTZAT), ATO	1
USAPRDC	1
DCSLOG	2
Rsch Anal Corp	1
ARO, Durham	2
OCRD, DA	1
USATMC Nav Coord Ofc	1
NATC	2
CRD, Earth Scn Div	1
USAAVNS, CDO	1
DCSOPS	1
OrdBd	1
USAQMCDA	1
QMFSA	1
CECDA	1
USATB	1
USATCDA	1
USATMC	20
USATC&FE	4
USATSCH	3
USATRECOM	17
USATTCA	1
USA Tri-Ser Proj Off	1
TCLO, USAABELCTBD	1
USASRDL LO, USCONARC	2
USATTCP	1
OUSARMA	1
USATRECOM LO, USARDG (EUR)	1

USAEWES	1
TCLO, USAAVNS	1
USATDS	5
USARPAC	1
EUSA	1
USARYIS/IX CORPS	2
USATAJ	6
USARHAW	3
ALFSEE	2
USACOMZEUR	3
USARCARIB	4
AFSC (SCS-3)	1
APGC (PGAPI)	1
Air Univ Lib	1
AFSC (Aero Sys Div)	2
ASD (ASRMPT)	1
CNO	1
ONR	3
BUWEPS, DN	5
ACRD(OW), DN	1
BUY&D, DN	1
USNPGSCH	1
CMC	1
MCLFDC	1
MCEC	1
MCLO, USATSCH	1
USCG	1
NAFEC	3
Langley Rsch Cen, NASA	2
Geo C. Marshall Sp Flt Cen, NASA	1
MSC, NASA	1
Ames Rsch Cen, NASA	2
Lewis Rsch Cen, NASA	1
Sci & Tech Info Fac	1
USGPO	1
ASTIA	10
ASD, FCL	1
HumRRO	2
US Patent Ofc, Scn Lib	1
USAMOCOM	3
USSTRICOM	1
USAMC	1
Hughes Tool Co	10

Hughes Tool Company - Aircraft Division, Culver City, California, HOT CYCLE ROTOR DUCT CLOSURE VALVE SYSTEM, TCREC Technical Rept 62-103, March 1963, 85 pp. (Contract DA 44-177-TC-832) USATRECOM Task 9R38-01-020-03.

Unclassified Report

A detailed analysis of the design and operation of a hot cycle rotor duct closure valve system has been

(over)

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2. Contract DA 44-177-TC-832

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